MECHANISMS FOR INTERMITTENT MOTION

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To Anne, David, Peter, and Leila for patiently allowing me the time and privacy required for a project of this magnitude.

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MECHANISMS FOR INTERMITTENT MOTION

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Glossary

ACCELERATION. The rate-of-change of the velocity of a body as a function of time. A vector quantity.

Angular. The rate-of-change of angular velocity of a body as a function of time.

Linear. The rate-of-change of linear velocity of a body as a function of time.

of gravity. Linear acceleration imparted to a falling body by the earth's gravitation field: equals approximately 32 ft per second^2 in English units.

ASYNCHRONOUS. Running at a variable rate or frequency. For example, a mechanism whose output is produced infrequently, perhaps only on command of the operator, is said to be running asynchronously.

BODY. A piece or portion of a mechanism or machine as distinct from the remainder of the machine.

Elastic. A body whose dimensions change and which typically absorbs and retains and/or releases energy while being used.

Free. A piece or portion of a machine that has been theoretically isolated from the remainder for analysis.

Rigid. A body whose dimensions are assumed not to change in use.

BRAKE. A mechanism or device whose function is to grab and stop a moving body upon command.

CAM. A rotating or sliding machine member whose function is to impart a predetermined motion of some sort to a second part called a cam follower that rolls or slides along a surface of the cam.

CYCLING RATE. The rate or frequency with which output motions are produced by an intermittent motion mechanism. Steps per minute.

CLUTCH. A mechanism or device whose function is to couple one machine member (usually a rotating shaft) to another (usually a second shaft) upon command.

COEFFICIENT OF RESTITUTION. An experimental constant used in the study of impact. It is a measure of the amount of energy lost during a collision between two bodies. A coefficient of 1.0 indicates that no energy is lost. The coefficient of 0 indicates that all energy is lost.

DAMPER. A mechanism or device whose function is to suppress the vibrations induced in a machine or system.

DAMPING CONSTANT. An experimental constant which gives a measure of the degree to which a damper will suppress vibrations in a given machine or system.

Critical. A damper is said to have a critical damping constant if vibrations in the machine or system are suppressed after the first half cycle of displacement.

DIFFERENTIATION. A mathematical or graphical process of determining the rate-of-change of a variable. If we differentiate a curve which represents the velocity of a body as a function of time, for example, we determine the rate-of-change of velocity at various times and hence, the acceleration of that body at those times. The process of differentiation can be performed mathematically, graphically, or experimentally.

DISPLACEMENT. The change in position of a body as a function of time.

Angular. The change in rotational position of a body as a function of time.

Linear. The change in linear position of a body as a function of time.

DWELL. Pause—Rest. An intermittent motion mechanism that is producing no output motion is said to be dwelling.

DWELL-MOTION RATIO. The ratio between the length of time occupied by dwell and the length of time occupied by output motion for an intermittent motion mechanism. The ratio would be very high for a mechanism that produces brief motion after a long rest period. The ratio would be very low for a mechanism that produces a great deal of motion with only short rest periods in-between each motion cycle.

ESCAPEMENT. A mechanism or device that is designed to alternately grab and release a rotating shaft, wheel, slide, etc.

A clock and watch. An escapement designed especially for use in a mechanical or electro-mechanical timekeeper.

Inverse. A mechanism or device which has geometrical similarity to a true escapement, but which provides the source of power which moves the output shaft rather than just controlling power received from some other source. Not a true escapement.

Machine. An escapement designed to control a rotating shaft, wheel, slide, etc., in a machine instrument or mechanism other than a timekeeper.

FORCE. The phenomenon which causes the acceleration of a body. Forces can be generated by fields (electrical, magnetic, gravitational) or by interactions between bodies.

GEAR. A machine member, generally circular though sometimes having an elliptical or other noncircular shape, whose active surface is provided with teeth that typically engage a similar gear to impart rotation from one shaft to another.

Cycloidal. A special family of gear systems in which one gear essentially rolls along the surface of another, rather than just rotating on fixed centers.

Differential. A special class of cycloidal gearing that is usually used to add or subtract the rotation of one shaft to or from another.

Mutilated. A gear on which some of the teeth have been purposely removed to modify the performance characteristics of a particular system.

GENEVA. A special type of machine or instrument cam that is frequently shaped something like a Maltese cross.

Instrument. Small lightweight Geneva intended for use in instruments and other systems where load requirements are light.

Machine. Heavy-duty Geneva.

GRAPH. A pictorial representation of a mathematical or experimental function. For example, we can draw a graph of the displacement of a body as a function of time.

INDEXING MECHANISM. Generally a mechanism or device that is designed to produce intermittent motion. In machine tool parlance, however, an indexer is usually a mechanism or device used to move an in-process or finished body from one work station to another.

INDEXING PRECISION. The degree of accuracy with which an intermittent motion mechanism or device positions the load.

INERTIA. A measure of the resistance of a body to linear acceleration. A function of the mass of the body.

INPUT. The power input to an intermittent motion mechanism or device. In some cases this will take the form of a rotating shaft driven by an electrical motor (for example in most Geneva, gear, or cam systems). In other cases, the input will be electrical and will be converted to mechanical motion by a solenoid or stepping motor.

INSTANT CENTER. The point about which a body is rotating at one instant in time. For a wheel rotating on a shaft, the instant center is the center of the shaft and does not move. For a wheel rolling along the ground, the instant center is the point of contact between wheel and ground and changes continuously.

INTEGRATION. The process of finding the sum or total of a large number of small values. The act of finding the area under a regular or irregular curve on a graph. Integration can be performed mathematically, graphically, or experimentally.

INTERMITTENT MOTION MECHANISM. Mechanism or device designed to produce intermittent output motion. Input can be either intermittent or continuous.

JERK. The rate of change of the acceleration of a body as a function of time.

LOAD CAPACITY (RELATIVE). The capacity of an intermittent motion mechanism to drive a load. If a body has a high relative load capacity it is able to move and control a very large load.

MASS. A quantity of matter.

MECHANICAL ADVANTAGE. A mechanism or device is said to possess mechanical advantage if a small input force or torque can produce a large output force or torque. A lever can have mechanical advantage, for example.

MOMENT OF INERTIA. A measure of the resistance of a free body to angular acceleration. A function of the mass and geometry of the body.

MOTION CURVES. Graphical representation of the displacement, velocity, and acceleration of a body as a function of time.

OUTPUT. The mechanical output of an intermittent motion mechanism. Consists generally of motion periods followed by dwell periods, followed by motion periods, etc.

PRIM. Programmed Rotary Intermittent Motion mechanism. A name given to a particular type of intermittent motion mechanism by its inventor. (See Figs. 15–25 through 15–27.)

RATCHET. A class of intermittent motion mechanism in which the input device is generally an oscillating tooth or pawl of some type, operating against a gearlike cam called a ratchet wheel.

Cam ratchet. A ratchet in which the basic input to the driver is provided by a rotating cam.

Impulse ratchet. A ratchet mechanism in which the input is provided by electrical pulses to a solenoid or equivalent.

RELIABILITY. A measure of the probability that a mechanism or device will perform its intended function correctly.

SLOPE. The rate-of-rise (or fall) of a mathematical curve. Equals the tangent of the angle between a line drawn tangent to the curve at the point whose slope is to be determined and the horizontal axis (see Fig. 2–10).

SPRING CONSTANT. A measure of the stiffness of a spring. The spring constant is low if it takes little force to extend a spring a considerable distance.

STABILITY. A measure of the degree to which an intermittent motion mechanism or device will continue to produce the desired pattern of output motion over a long period of time. Related to reliability but more a measure of the degree to which an output motion will change, than as to whether or not it will be produced at all.

STAR WHEEL. A particular class of intermittent motion mechanism that has some of the properties of a Geneva and some of a mutilated gear (see Figs. 15–2 through 15–7).

STEPPING MOTOR. An electrical motor designed to produce intermittent rather than continuous output.

SYNCHRONOUS. Fixed rate or frequency. A synchronous motor, for example, will rotate an output shaft at a constant angular velocity in spite of reasonable variations in such things as input voltage, output load, etc. A Geneva mechanism driven by a constant velocity rotating shaft will produce output motions at a fixed frequency and thus could be said to produce a synchronous output.

TORQUE. The product of an applied force and the radius through which it is acting about a center of rotation.

TRANSIENT. Changing, temporary, passing from one state to another.

VECTOR. A mathematical invention used to describe the properties of a quantity when both magnitude and direction are involved. Force, velocity, and acceleration can all be described by vectors.

VELOCITY. The rate-of-change of displacement of a body as a function of time. A vector quantity.

Angular. The rate-of-change of angular displacement of a body as a function of time.

Linear. The rate-of-change of linear displacement of a body as a function of time.

VIBRATION. The periodic motion of a body or portion of a body in alternating and opposite directions from a position of equilibrium.

Preface

Intermittent motion mechanisms play such an important role in modern technology that we might say pacifists would make the world far "safer" by banning intermittent motion than by banning the bomb. Intermittent motion mechanisms are essential elements of such machines as paper tape and card punching equipment, high-speed typewriters, digital servo systems, mechanical and electromechanical counting apparatus, automatic weapons, clocks and watches, and production and assembly machines of all sorts. Nor is the end in sight. Although electronics are certainly going to take over most control and computing functions, man will long have need for "muscle" as well as "brain" and intermittent motion devices will continue to play a significant role.

It is very useful, therefore, for machine and instrument designers to know something about the mechanics and mechanisms of intermittent motion, yet it is rare to find a designer who really understands such devices even if his industry makes use of them. The reason is that intermittent motion is not adequately covered by most engineering texts on mechanics, machine design and the like. Those texts that do cover it are so complicated and mathematical that they are beyond the level of most practicing designers. Furthermore, their authors usually end by inferring that, "exact solutions to these design problems are impossible."

The basic problem is that intermittent motion is a "special case" in several ways:

1. It involves transient mechanics rather than steady-state mechanics.
2. Because impact is often involved, wear characteristics differ somewhat from those found in machines where only sliding contacts are found.
3. Impact and sudden changes in acceleration and velocity lead to force and stress levels that far exceed the apparent (static) force and stress levels that could be produced by the motors, springs, and other power plants the designer has provided.
4. High stress levels, backlash, vibration and resiliency of parts lead to significant control and stability problems that are not found in steady-state machinery.
5. When high stress levels are involved, machine elements and linkages can no longer be considered as rigid bodies, thereby making it impossible for the designer to use much of the mechanics which he was originally taught.

In this text we will try to give the designer insight into some of these things. Like most texts in elastic body and intermittent motion mechanics we will end up by concluding that we cannot obtain rigorous numerical solutions for our design

problems. Unlike most such texts, however, we hope to discuss these problems on a level which will be intelligible to the average designer and thereby give him at least a better understanding of intermittent motion mechanisms and their special problems, so that he can design them to maximize favorable features and minimize unfavorable features.

As a matter of fact, any mechanism or machine with clearance and backlash has some of the problems of an intermittent motion mechanism. Since it is almost impossible to build a machine without clearance or backlash somewhere, one might say that all designers have these problems, usually without realizing it or the implications.

A Text for Designers

This text is not intended for physicists, but for graduate engineers or for non-graduates who have some formal training in basic mechanics. We assume they will know, for example, what a force is, or a mass, and can tell the difference between acceleration and velocity. We will, however, review some of these things, using graphical and pictorial methods instead of mathematics, before getting into a discussion of the complexities of intermittent motion. This is not a complete course in graphical mechanics; only those topics which are pertinent to the purposes of this text will be covered. We hope, however, that by using the graphical approach most readers will gain an insight and understanding into intermittent motion that they would not obtain through the complex mathematical approach.

Like most technical authors, I am indebted to many people for the material which appears in my book. Principal among my own teachers were Francis W. Sears, now at Dartmouth; Dr. Igor Paul, of M.I.T.; Dr. Will Roth of the University of Vermont; and Dr. Karl Maier of Math & Metric, Inc.; all of whom struggled at one time or another to educate me.

I also have to thank, most sincerely, my employers at the Veeder-Root Company in Hartford. Our mutual struggles to make ever better counters (intermittent motion mechanisms) have provided most of the time and resources necessary for my education in this subject.

I must also thank the many individuals, publishers and manufacturing companies who gave me information on, and permission to publish illustrations of their various products and inventions. Special thanks, is due the Penton Publishing Company, publishers of *Machine Design* magazine, for permission to reprint the discussion of impact in Chapter 3. I originally wrote this as an article for their magazine. Penton has also given me permission to reuse many illustrations I drew for articles on several different types of intermittent motion mechanisms; articles which also appeared in *Machine Design*.

The McGraw Hill Publishing Company has also generously allowed me to use a large number of drawings which originally appeared in their *Product Engineering* magazine as illustrations for articles by many different authors.

My colleague, Mr. John Gamble, also deserves much credit for his invaluable assistance on the chapter on stepping motors. Other reviewers to whom I am indebted include: Mr. George Coxeter of Raymond Engineering Company; Mr. John Murray of the Superior Electric Company; Mr. Dennis Klipp of the Commercial Cam and Machine Company; Mr. Martin Zugel of Cyclo-Index Corporation; Mr. Peter Jackson of Jackson Automation; and Mr. H. S. Cummings Jr., of the Lowell Corporation, each of them has reviewed portions of the text and made valuable suggestions.

Finally, thanks are due to Earline Ashley for preparing the manuscript. As a two-finger typist, I never would have made it alone!

CHAPTER 1

Introduction and Some Basic Concepts of Mechanics

INTRODUCTION

Why We Need "Theories"

To design, evaluate, or understand any kind of mechanism it is helpful to be able to measure or predict the forces, accelerations, velocities, and displacements which occur in various parts of the mechanism. From this information the designer can then determine stress levels and performance characteristics. Will the mechanism do the job expected of it? Will it do it in the time available? Will it have the desired operating life, etc.?

But Theories Do Not Always Work

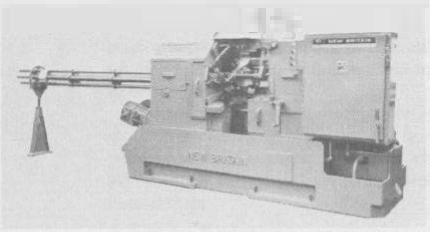
The engineer or designer acquires his understanding of force, acceleration, velocity, and displacement through his study of mechanics. Too often, however, his early struggles to apply his knowledge of mechanics to actual design problems defeat him. The abstract language of mathematics has not given him sufficient "feel" for the subject to allow him to apply it to a specific design situation, or the machine just does not seem to obey the rules which he was taught because the machine is not a collection of rigid bodies connected by ideal joints. It is, instead, a series of elastic bodies interconnected by links that introduce friction, backlash, and clearance, all subject to impact, vibration, and chatter; and it seems often to have a mind of its own and be determined never to complete the performance or life intended for it.

At this point, most beginners give up, usually decide that the theoretical approach is wrong, and that only a trial and error experimental approach gives the "correct" answer. Too frequently the result is a machine that works, but whose performance is so dependent on a delicate balance of forces and dimensions that the designer's employer has a long string of production problems ahead to contend with. Small variations in part dimensions, hardness, or material composition can lead to large and mysterious changes in machine performance and life.

Experimental Versus Theoretical Models

It would be far better if the designer had enough feel for, and patience with, theoretical design so that he could build a crude theoretical model of his machine in parallel with the construction of his experimental model. He should realize that both models are imperfect at the start—that both will require debugging and development. Thus, the theoretical model must be refined, when experiments with an actual prototype machine show that the theoretical model is inadequate, to describe the actual performance.

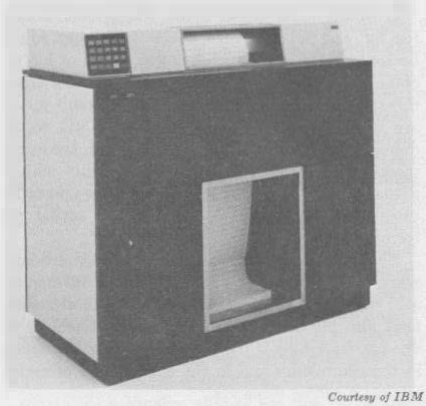
The physical model must be refined and debugged when its performance turns out to be inadequate, and particularly when the theoretical model shows that the performance of the actual machine will change drastically if there are slight variations made in the dimensions and in the composition of certain critical machine elements.



Courtesy of the New Britain Machine Company, Division of Litton Industries

Fig. 1–1. Multiple-spindle automatic bar machine uses a Geneva indexing mechanism.

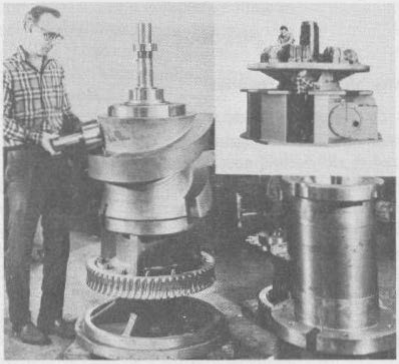
The designer who succeeds in simultaneously developing both a theoretical and physical model of a ne[illegible] reach the product[illegible] phase of the design process in full control of his new creation. He will be able t[illegible] the implicat[illegible] of dimensional, material, or configurational changes requested by tooling or manufacturing people without having to build a multiplicity of new models and subject them to life and performance tests. He will know which design tolerances to fight for and which to relax. He will know whether or not the sales department can [illegible] an important customer some modification in the published performance specifi[illegible]ion or whether such modificatio[illegible] clearly impractical or even impossible. Figures 1–1, 1–2, and 1–3 show substantial end products of the successful [illegible]



Courtesy of IBM

Fig. 1–2. High-speed [illegible] machine uses a variety of intermittent motion mechanisms

Some engineers and designers now are able to do [illegible] the [illegible]ly [illegible]ools [illegible] they were taught [illegible] standard courses on [illegible] design. In my experience however, the majority never reach th[illegible] happy state of aff[illegible], [illegible] because the ma[illegible] too abstract, or because the mathematics required for an actual design situation is, perhaps, too complex. By the time the designer has considered all the factors, his mathematical model may be so complicated that it can only be handled by large computer techniques. This is an e[illegible]ive, ti[illegible] process and only [illegible] sit[illegible]ations where huma[illegible] depend on the results



Courtesy of Ferguson Machine Company

Fig. 1–3. Large cam used to drive a loaded indexing table weighing over 13,000 lbs, 45 degrees in ¾ second.

(for example, in the aircraft industry) will the typical design boss tolerate the work required for such an [illegible]

[illegible]

It is also possible, however, to build a rough, graphical theoretical model of a new design, rather than a mathematical one. This approach has several things to recommend it. First of all, you use basically one technique (plot a curve) to solve all types of problems, rather than a mathematical approach where you handle each type of equation, i.e., linear motion, harmonic motion, damped harmonic motion with shock [illegible]xcit[illegible]ion, etc., differently. With graphics

you always follow the same step-by-step procedure regardless of the simplicity or complexity of the original data.

The graphical approach is started with a measurement or prediction of one of four quantities: force, acceleration, velocity, or displacement. It might, for example, be decided that the machine is to move through a certain pattern of motion in a certain time. This would provide a prediction of the displacement of the machine as a function of time. Simple graphical techniques are now used to estimate the velocity, acceleration, and forces which the machine must experience to perform the desired displacement cycle.

It should be emphasized that this is only a sketch-pad procedure. It does not lend itself to exact answers unless graphs and charts are drawn with great precision. At first glance, then, it would appear that graphical models suffer in comparison with mathematical models, but I maintain that in most machine design situations, mathematical models do not produce exact results either. They can produce exact results only when they consider *all* factors and have exact inputs, but this is hardly ever possible in a real design situation. The mathematical model, then, can only give approximate answers or indicate trends as changes in dimensions of a design are explored. A graphical model can do this also, and, I think, do it more readily. Its obvious visual message is far more enlightening to most designers than a page full of formulas and a table of numerical results.

The graphical approach to mechanics, furthermore, is ideal for a study of intermittent motion mechanisms, because here an exact mathematical model is not only difficult to build, it is often virtually impossible. There are too many unknowns, too many variables, to allow anything but an approximate solution for certain special and highly simplified cases of little interest to a practical designer. The graphical approach, on the other hand, bolstered by certain fairly easy-to-acquire experimental data, can show the designer approximately where he stands and in approximately which direction he should change his design to improve its performance and production life.

SOME BASIC CONCEPTS OF MECHANICS

Before we tackle graphical mechanics in a serious way, we must understand a few basic concepts such as force, velocity, angular acceleration, etc., and

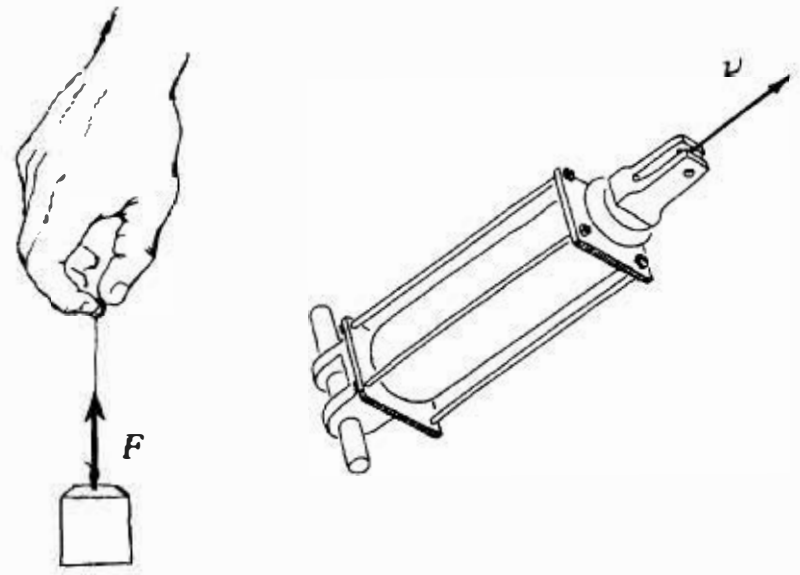


Fig. 1–4. Force and linear velocity vectors.

learn how to relate these to actual machine parts. This discussion will not be complete in a textbook sense; it is not intended to be a course in mechanics. It is expected that the reader will have had such a course and will use the material given here merely to refresh his understanding of some of the elementary ideas of mechanics needed in the discussion of intermittent motion and its mechanisms.

Vector Representation of Force and Velocity

Many things can be represented by *vectors*, which are mathematical quantities representing direction and magnitude, and are usually illustrated as small arrows. Linear and angular displacement; velocity and acceleration; force; torque; momentum; and impulse are all vector quantities. We will be concerned, however, with vector representation of only two of these: force and linear velocity. We will be using several of the others; angular velocity and torque, for example, but will be interested only with their scalar properties; that is, we will be concerned only with their magnitude and not with their vectorial *direction*. Figure 1–4 shows the vector representation of the force exerted on a block by a man, and the vector representation of the linear velocity of the piston of an hydraulic cylinder. Both vectors are shown as arrows whose directions indicate the direction of the force and velocity, respectively, and whose lengths represent the magnitude of those two parameters.

Resultants of Two Vectors

It is possible to add two or more vectors together to find what is called a *resultant vector*; a single

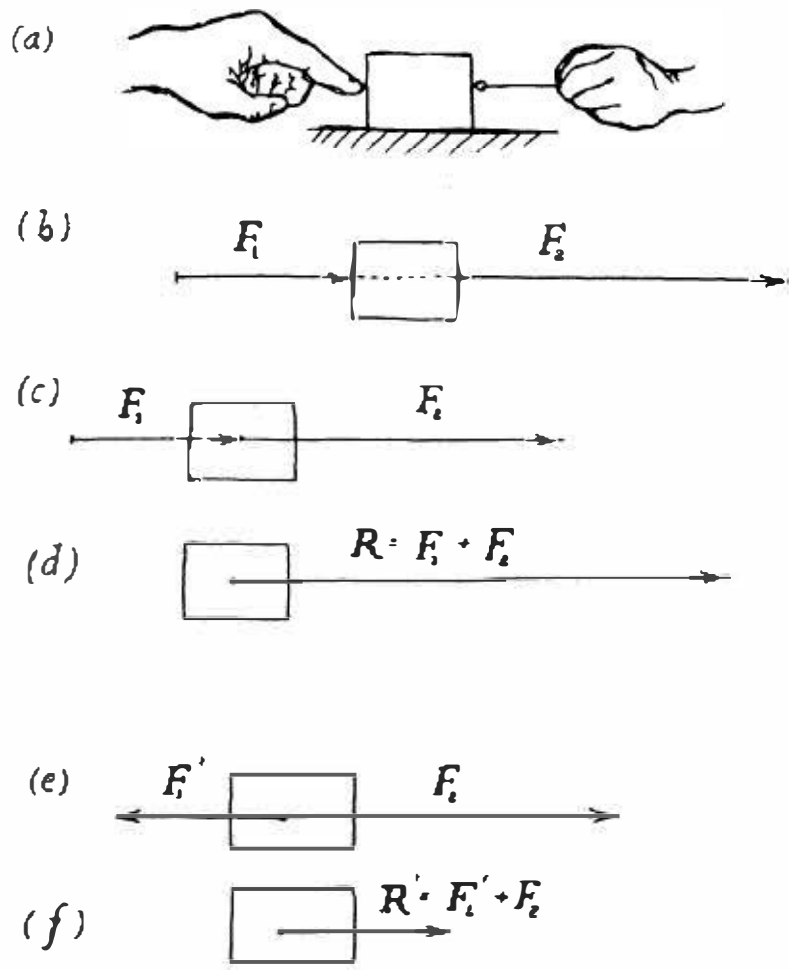


Fig. 1–5. Finding the resultant of co-linear vectors.

vector that combines or summarizes all the properties of the original group. In Fig. 1–5a, for example, one man is pushing on a block while a second is pulling on it. The two forces exerted on the block are shown in vector representation in Fig. 1–5b. Since both of these forces are: Fig. 1–5b, acting along the same line, and, Fig. 1–5c, pass through the same point in the block, they can be added together, Fig. 1–5d, by drawing a single vector, equal in length to the sum of the original two, and pointing in the same direction as the original two. If one of the two men pushed (or pulled) with a force of this magnitude and direction he would produce the same effects on the block as the two men produced in the original situation.

If both men pulled on the block in opposite directions, Fig. 1–5e, instead of one pushing and one pulling in the same direction, the resultant vector would be the difference between the two original vectors, and would be in a direction determined by the stronger man, as shown in Fig. 1–5f.

Vectors need not be acting along the same line to be added. In Fig. 1–6 we show the steps required to add two force vectors that are acting through a common point on a block but are not acting along a common line. Assume that the block is falling under the influence of its weight W, which acts through the center of gravity cg, as shown in Fig. 1–6a. This produces one vector force on the block in a vertical downward direction. As the block falls, a second force F, is exerted on the upper right-hand corner, as shown in the illustration. Notice that the line of action of force F, also passes through the center of gravity of the block. This is very important. The type of vector addition we are about to perform cannot be done unless all the forces involved have lines of action that pass through a common point.

Since these forces do have lines of action that pass through a common point, the center of gravity, we can add them together as shown in Figs. 1–6b and 1–6c. F is first "moved" along its line of action until its "tail" coincides with the tail of the vector W (Fig. 1–6b). A vector parallelogram is then constructed to find the resultant R. The net, or total force on the block is then represented by vector R, alone. The block would tend to move downward and to the left, as a result of the action of these two forces.

Again, this is just a refresher course on vectors; we assume the reader has already been exposed to these topics. If not, he is urged to consult a mechanics book if he cannot understand or remember the principles discussed here.

It is also possible to find the resultant of more than two vectors, or to find the resultant of two or more vectors whose lines of action do not pass through the same point, but these constructions will not be necessary for the purpose of this book.

Everything that has been said here about force vectors, of course, applies to linear velocity vectors as well. They can also be added, subtracted, etc., in the same way.

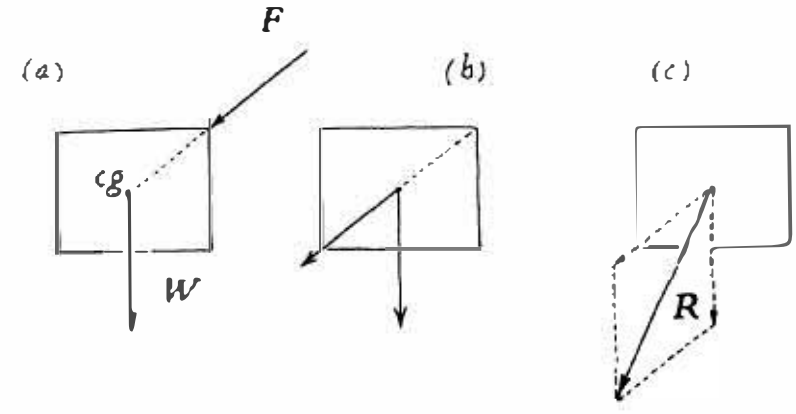


Fig. 1–6. Finding the resultant of two vectors whose lines of action pass through a common point.

Components of Vectors

We can reverse the process described above and convert a resultant, or single, vector into two or more components whose combined effect is equivalent to that produced by the original single vector quantity. This is frequently helpful in determining forces or velocities in certain directions that are of special interest in a given design problem. The component vectors do not have to be at right angles to each other. In Fig. 1–7, for example, five possible ways of breaking a single vector down into two components are shown. An infinite number of component pairs are possible and all are equally correct. It is only necessary that the original vector be the diagonal of the parallelogram defined by the two component vectors that replace it.

Again, the purpose of doing this is to determine what is happening in directions of special interest in a given design situation. In Fig. 1–7e, for example, the heavy vector (original resultant) describes the instantaneous linear velocity of a drive pin. This velocity vector has been broken down, for study

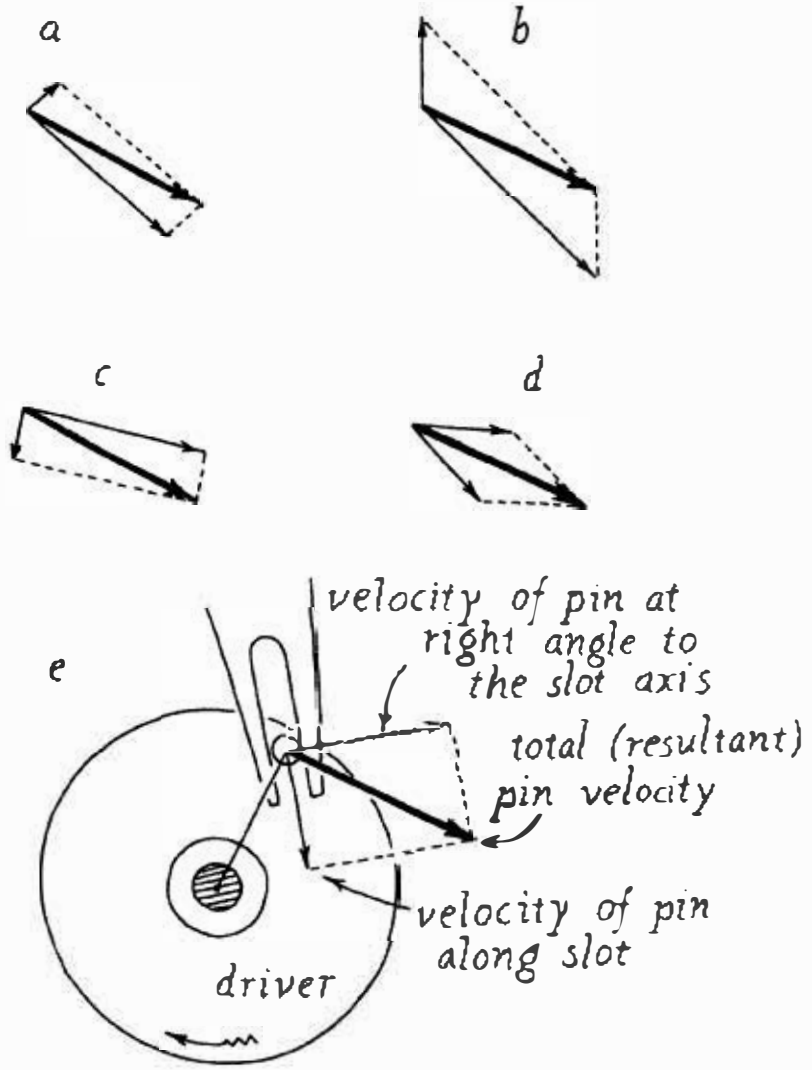


Fig. 1–7. Five ways to divide the same vector into two components. Each of these groups is correct, and there are many other available choices.

purposes, into two components; one showing the instantaneous velocity of the pin along the axis of the slot; the second showing the instantaneous velocity of the pin at right angles to the slot. As we will see later, the ability to determine vector components such as these is essential if we are to study the various velocities of different parts of a machine.

Free Bodies

Whenever we use vectors to describe the forces acting in a given design we must be careful not to confuse the forces acting *on* a body with the forces produced *by* the body on something else. To avoid this difficulty we make use of the concept of the *free body*, which can be either a single machine part or an assembly of parts, but which, in every case, is treated as a single item. In Fig. 1–8a, for example, an intermittent motion mechanism is shown, which consists of a heart cam, a spring and solenoid drive system, and a drive linkage. At the moment shown in the picture the solenoid has been actuated and the linkage and cam are moving in the directions shown by the small arrows with broken tails. (These arrows will be used throughout the text to show direction of motion; they are not vectors as they show direction only.) In Fig. 1–8b, c, d, and e are shown the heart cam, the upper drive roller, one bar of the linkage and the rest of the system, as four separate free bodies. Shown in each case are the forces acting *on* that free body only; never the forces produced *by* that body on something else. We could have treated the other linkage parts, the solenoid, the spring, etc., as separate free bodies instead of the assembly shown in Fig. 1–8e if we had wished.

Looking at Fig. 1–8b, we see the forces produced by the drive roller and the supporting shaft *on* the heart cam, which is the free body in this case. Each force is also accompanied by a small friction force represented by a vector at the base of the main vector and at right angles to it, since that will be the direction and relative magnitude of these friction forces. We could have shown each pair of forces, friction and main force, as a single resultant, of course, but chose to show them this way, since we are usually interested in the effects and directions of the friction forces separately.

The force produced by the roller on the cam is mirrored by the force produced by the cam on the roller as in Fig. 1–8c, where the roller is now the free body. Every force is accompanied by an equal

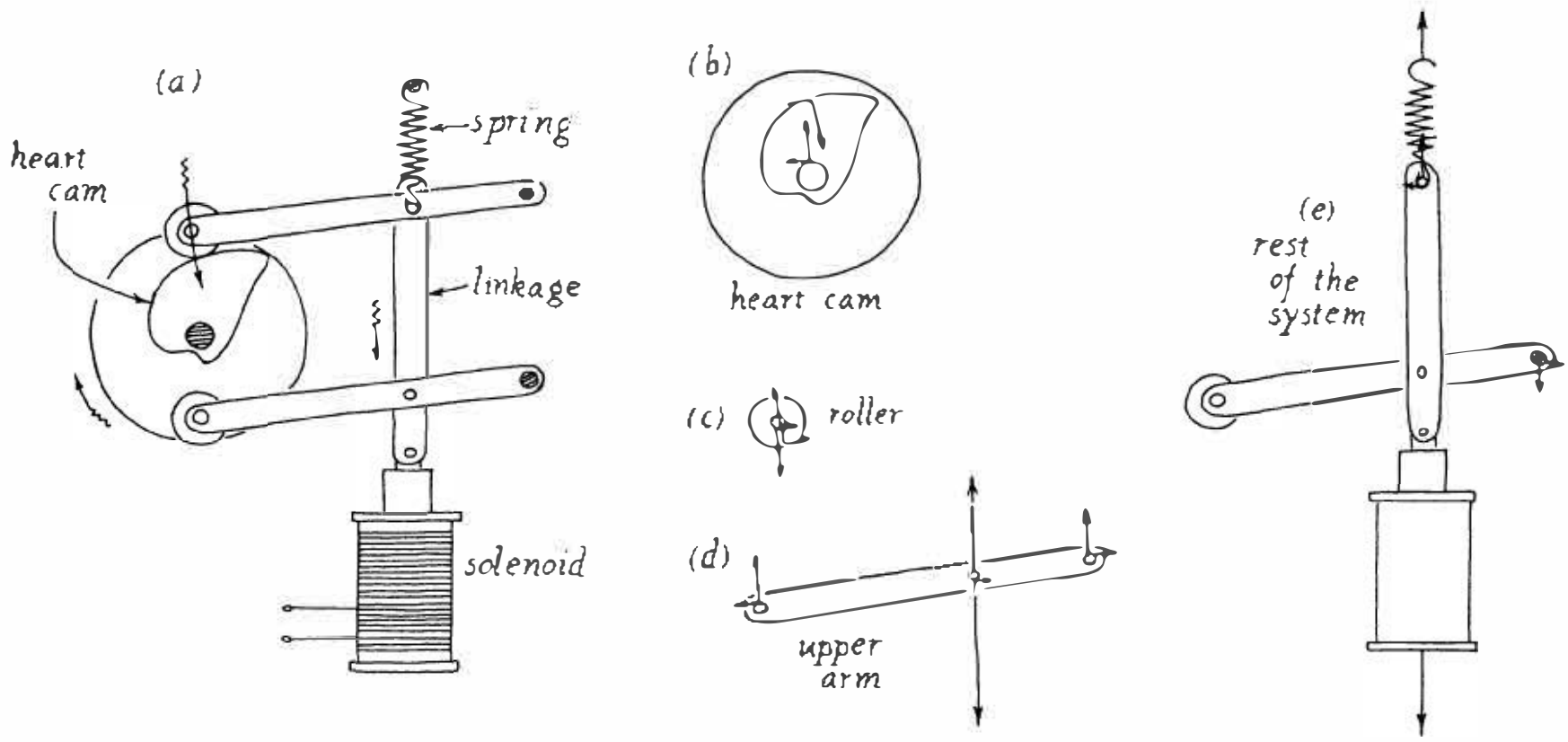


Fig. 1–8, Dividing an intermittent motion mechanism into free bodies for force or velocity analysis. The free bodies can consist of individual parts or groups of parts.

and opposite force somewhere in the system, as Newton taught us, and as you learned in your mechanics course. The drive roller is also acted upon by the upper arm.

The upper arm, in turn, is acted upon by the roller, the spring, the other parts of the linkage, and the upper-right pivot shaft. The rest of the system is acted upon by the table on which the solenoid is mounted, the pin which restrains the upper end of the spring, the upper arm, and the lower-right pivot shaft, as shown in Fig. 1–8e.

Again, in each case, we have shown the forces acting on the part or group of parts selected as the *free body*. We have never shown forces acting within the body, or produced *by* the body *on* something else.

Determining the Directions of Forces in Machine Bodies

Most designers could probably identify the locations of the forces acting on the various free bodies in Fig. 1–8, but many would have difficulty in determining the actual direction of these various forces. The direction of a force is not determined by the direction of motion of the body producing that force, but rather by the shapes of the bodies involved and the presence or absence of friction.

Shown in Fig. 1–9, for example, is the force exerted by a driver cam on a load cam at one instant in time. This force F_D is perpendicular to a line that is tangent to both surfaces at the point of contact. Notice that F_D is not tangent to the arc described by the instantaneous contact point of the driver, nor the arc described by the instantaneous contact point of the load. Instead, the direction of F_D is determined by the shapes of these two surfaces. In the illustration the two surfaces are assumed to be

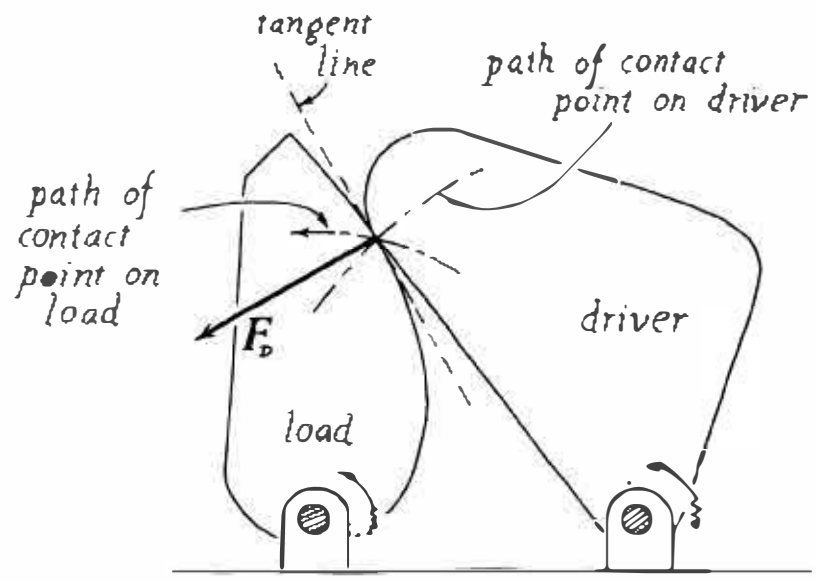


Fig. 1–9. Determining the direction of force exerted by a drive cam on a load cam, assuming that both cams have frictionless surfaces. The resultant force will be perpendicular to the cam surfaces at the point of contact, and its direction will not be influenced by the direction of motion of the contact points on the two cams.

frictionless and so the only force which can be produced by one cam on the other is a force normal to their surfaces at the point of contact.

It is also useful to consider the torque produced on the load by the driver in Fig. 1–9, and the torque required of the driver. These are not necessarily the same. Since torque is the product of force-times-radius (with the radius drawn perpendicular to the line of action of the force), we see from Fig. 1–10 that the torque generated by the driver is $F_D \times r_1$ and that the resulting torque produced on the load is $F_D \times r_2$ (units in both cases: ft-lbs). Assuming no friction, there is no loss of energy in the system. Due to the respective cam contours, the load will

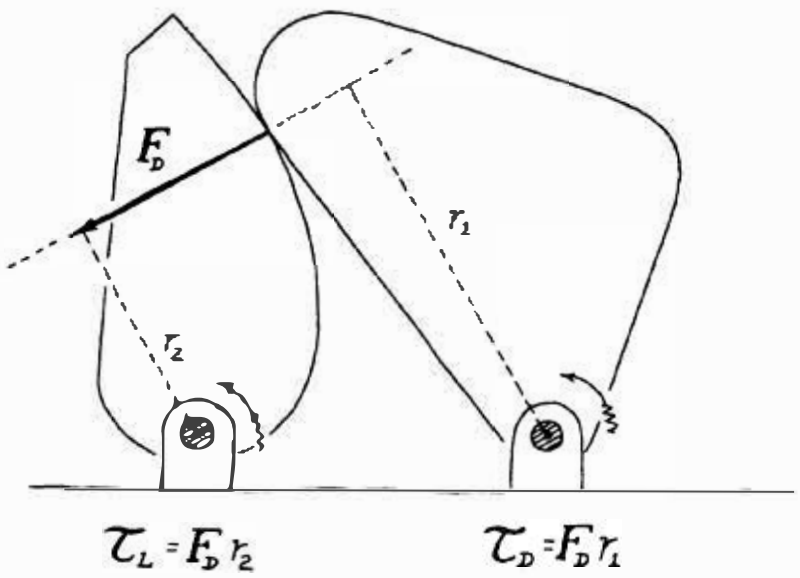


Fig. 1–10. Determining the torque required of the driver and produced by the driver on the load in a frictionless-drive situation.

move through a larger angle than the driver, during their operation, and the input work (torque times input displacement angle) will be equal to the output work (torque times output displacement angle).

Including Friction

Figure 1–11 shows the same pair of cams at the same instant of time, but we are now back in the real world and will consider the effect of the friction forces generated between the cams during their operation. The driver still produces a normal force F_D, on the load. This, in turn, produces a friction force F_F, as shown in Fig. 1–11. The direction of the friction force is determined by the direction of motion of the driver over the surface of the load cam. Remember that we are drawing the forces produced by the driver on the load. If you study the illustration

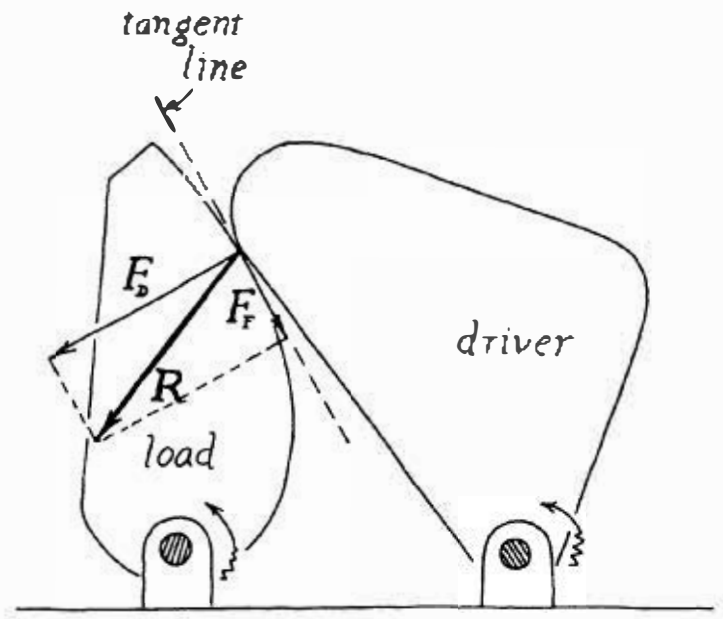


Fig. 1–11. Determining the direction of the resultant force produced by a drive cam on a load cam when friction is present.

a minute, you will see that as these two cams continue to turn, the driver will move down over the surface of the load cam, pulling or scraping on it in the direction shown by the friction force vector.

The resultant force produced by the driver on the load is the sum of the normal force F_D, and the friction force F_F, and is shown as resultant R in Fig. 1–11.

Calculations of drive torque and the torque produced on the load by the driver must also take the friction force into account. This is most easily done by basing the calculations on the resultant rather than on the friction and normal forces separately. In Fig. 1–12 we see that the torque on the load is

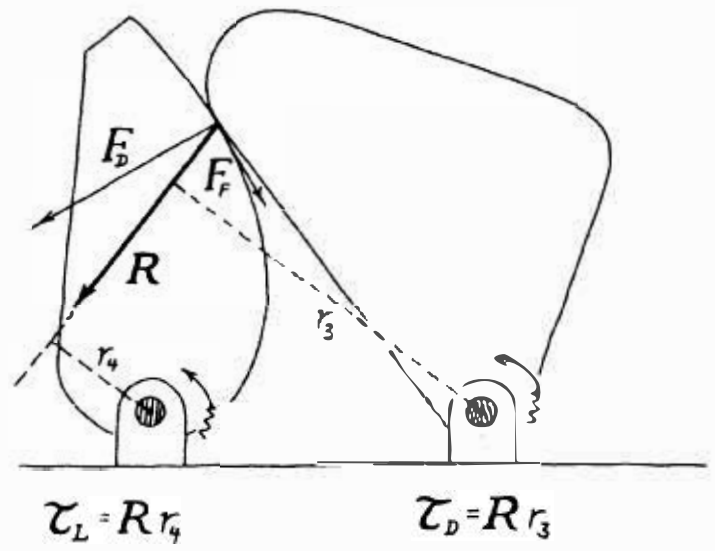


Fig. 1–12. Determining the torque required of the driver, and produced by the driver on the load when friction is present.

$R \times r_4$ and that the torque required of the driver is $R \times r_3$. (Again, the units for the torque are ft-lbs.)

Although net resultant R is slightly longer than the normal force F_D, you will find that the torque produced by the driver on the load, in this case, is less than that of the frictionless case discussed earlier, because r_4 in Fig. 1–12 is much smaller than r_3 in Fig. 1–10. The useful work transmitted to the load from the driver has been decreased by the friction, as we would expect. This is not always the case, however. It is possible to design a cam system in which the friction forces at the drive surface aid the drive force. So-called recess-action gears use this principle, for example.

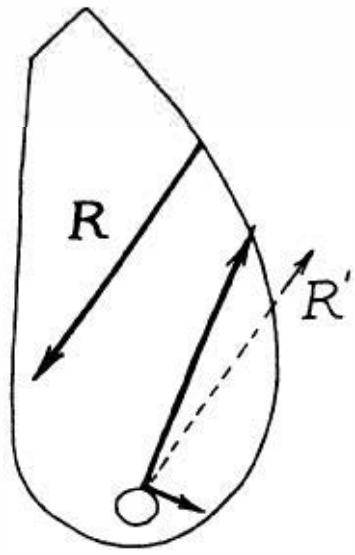


Fig. 1–13. Possible direction of pivot reaction forces on the load cam of the system shown in Fig. 1–12.

Direction of Pivot Forces

What about the direction of the forces produced by the pivot on the load cam in Fig. 1–12? How do we determine their direction? The answer again comes from basic mechanics. Remember the concept of equilibrium? If the net linear motion of a body in the horizontal direction is 0, as it would be in the case of a cam rotating about a pivot, then the net resultant of all the forces in the horizontal direction on the body must also be zero. Similarly, the forces in the vertical and perpendicular directions must be 0 if the body does not move in those directions either. From these facts we write "equations of motion" which will enable us to determine the so-called X, Y, and Z forces produced by the pivot on the body, and ultimately, the resultant force produced by the pivot on the body. The real-life

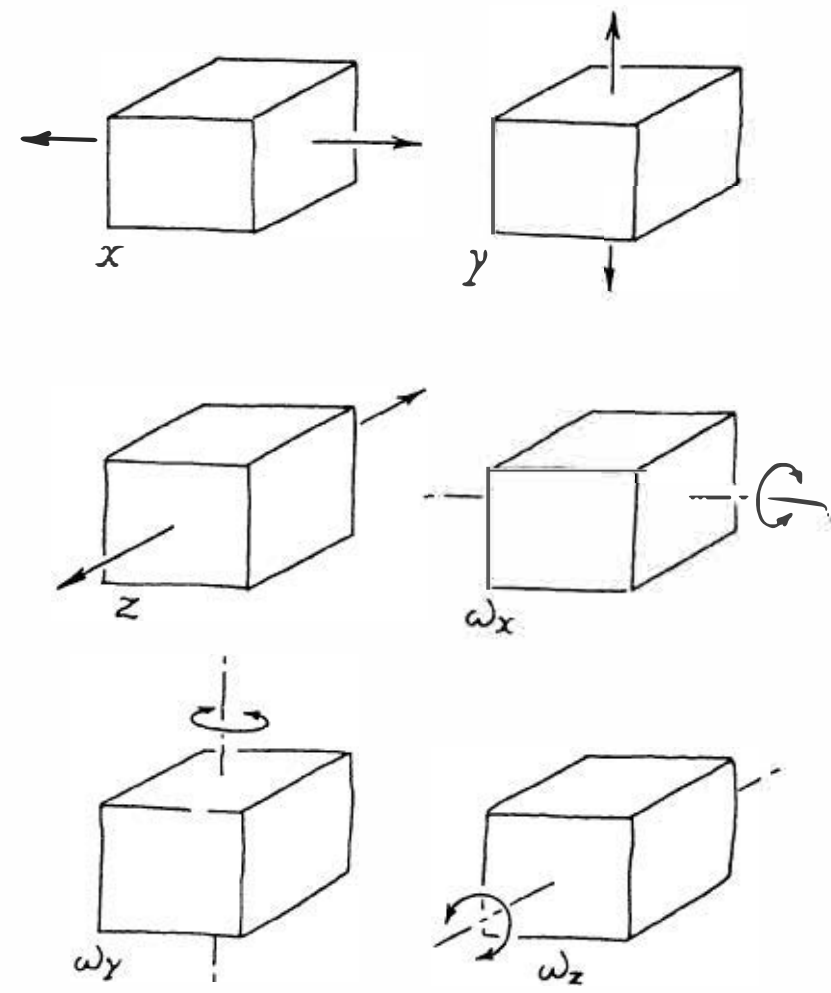


Fig. 1–14. A free body has six degrees of freedom; three in translation and three in rotation, as shown. Any set of mutually perpendicular axes could be chosen.

situation is complicated by the fact that the pivot force consists of a reaction or supporting force and a friction force, and that we usually want to know these separately. The equilibrium equations only give us a resultant. In many cases, we must assume frictionless pivots to avoid what is called a "statically indeterminate situation" and arrive at any conclusion at all!

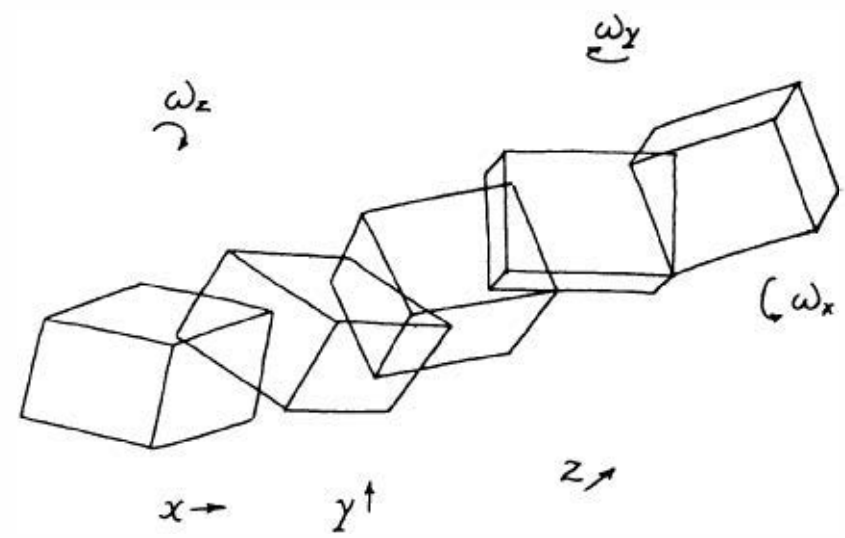


Fig. 1–15. A truly free body can have simultaneous motion in all six degrees of freedom.

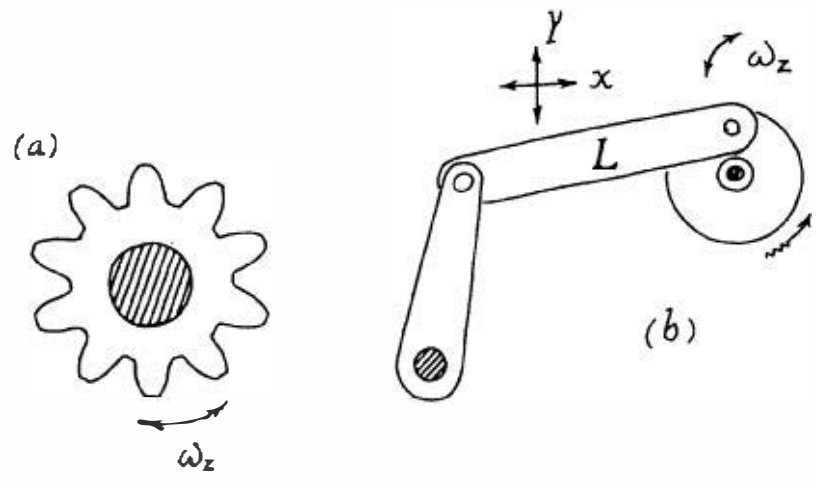
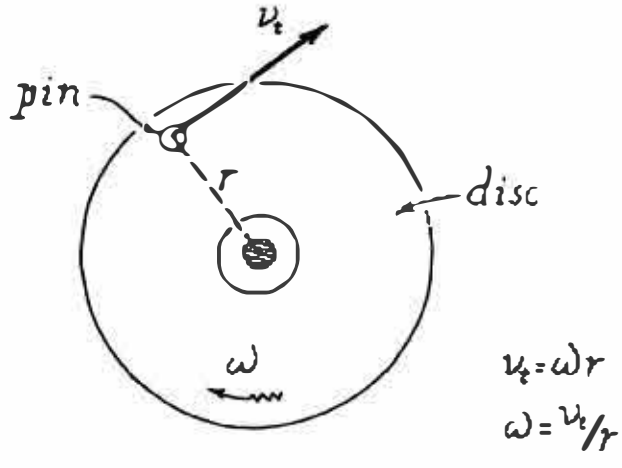


Fig. 1–16. Most machine bodies are constrained by bearings and structures and thus have limited numbers of degrees of freedom. The gear, above, can move in only one way. Linkage *L*, can move in three ways simultaneously, but only for a limited distance.

Fortunately, we will not need to get into all of this to understand intermittent motion mechanisms, nor will we be using equations of motion here. But the designer should be aware that pivots and supporting structures do, indeed, exert supporting and reactionary forces on bodies in response to those produced by the loads and driving members, and he will sometimes want to predict the approximate directions of such forces and reactions. Figure 1–13 shows a "guess" of the resultant pivot force R', and the normal and friction components of R', for the load cam of Fig. 1–12. If the discussion above is not a sufficient reminder for the designer, he must turn to his book on mechanics for a more detailed discussion.



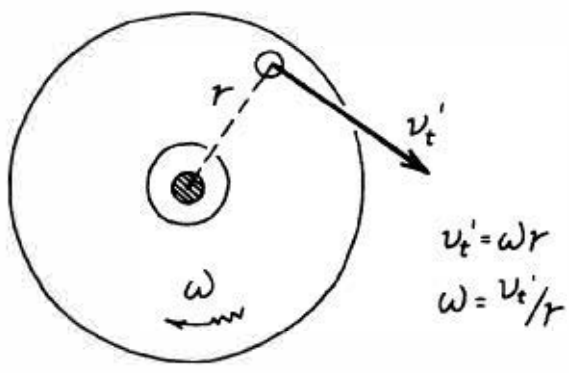


Fig. 1–17. Instantaneous tangential velocity of a pin on a disc. The disc is assumed to be rotating with a constant angular velocity, ω.

Degrees of Freedom

Every free body has what is called six degrees of freedom, as shown in Fig. 1–14. The body can have linear motion in the *X*, *Y*, or *Z* directions or it can rotate about the *X*, the *Y*, or the *Z* axes. The choice of these three axes is not fixed. You can choose any three directions that suit your problem, as long as they are mutually perpendicular.

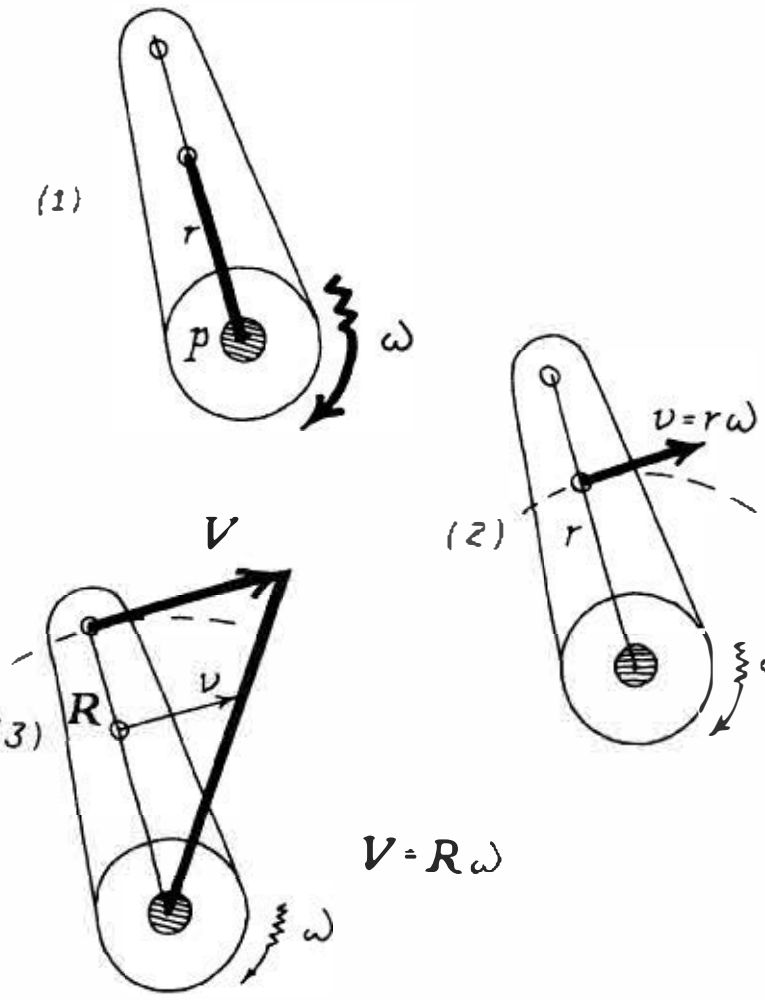


Fig. 1–18. Showing the relationship between the instantaneous tangential velocity of two pins located at different radii on a rotating link.

A truly free body can move in a very complex fashion, as suggested by Fig. 1–15. At first glance, this would appear to be motion not defined by any of the six degrees of freedom shown in Fig. 1–14. Mathematically, however, complex motion is merely

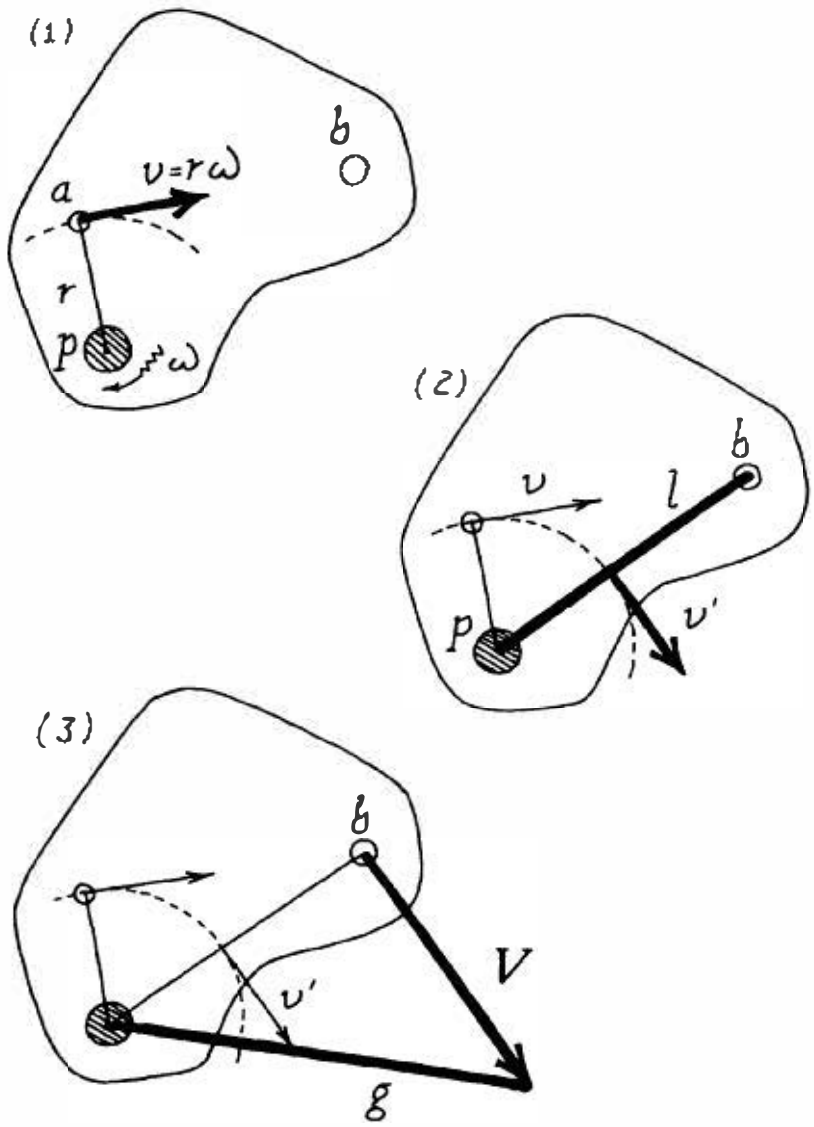


Fig. 1–19. Graphical construction to find the instantaneous linear velocity of pin *b*, on an irregular link, when the instantaneous linear velocity of pin *a*, is known. (Pin *a*, for example, might be driven by a crank whose velocity has previously been determined.)

(1) Known: v, r.

(2) Construct v' tangent to the arc (the dotted line describing the path of pin *a*), starting on line l, as shown. v' is equal to v, in length, since both are the same distance from center, p.

(3) Construct v parallel to v', using construction line, g. This is the same triangle construction used in Fig. 1–18.

a combination of two or more of the six basic motions allowed the body. In Fig. 1–15 the body is moving simultaneously in the X, Y, and Z directions, while rotating simultaneously about the X, Y, and Z axes.

In our analysis of machine motions, we will sometimes study the simultaneous motion of a machine link along or around more than one axis. Fortunately, however, machine bodies are always constrained to some extent, by bearings and supporting structure, and so can never (I suppose *never* is overly optimistic!) go through the gyrations suggested by Fig. 1–15. The gear in Fig. 1–16a, for example, can only rotate about the Z axis, which we assume is perpendicular to the page. It cannot translate (move linearly) along the X or Y axes without shearing the shaft on which it is mounted. It cannot move along that shaft (along the Z axis) without presumably breaking the pin or collar which holds it to the shaft. And it cannot rotate about the X or Y axes without breaking the shaft. Thus, we need only consider one degree of freedom to analyze its normal motion.

Link L, in Fig. 1–16b, is allowed a little more freedom of motion. It is permitted a little motion in the X and Y direction and can rotate through a small angle about the Z axis, but rotation about the X and Y axes and translation along the Z axis is prohibited by the way L is connected to the other members of the linkage train.

Velocities of Machine Members

The velocity of a body is officially defined as the rate-of-change of displacement of the body as a function of time. Velocity is a vector quantity, a measure of both magnitude and direction.

Linear velocity is relatively easy to understand and visualize. An automobile has a velocity of 60 miles per hour in a direction 23 degrees east of north, for example. Velocities associated with rotating parts are a little more difficult to understand, since both linear velocities (rate-of-change of linear displacement as a function of time) and angular velocities

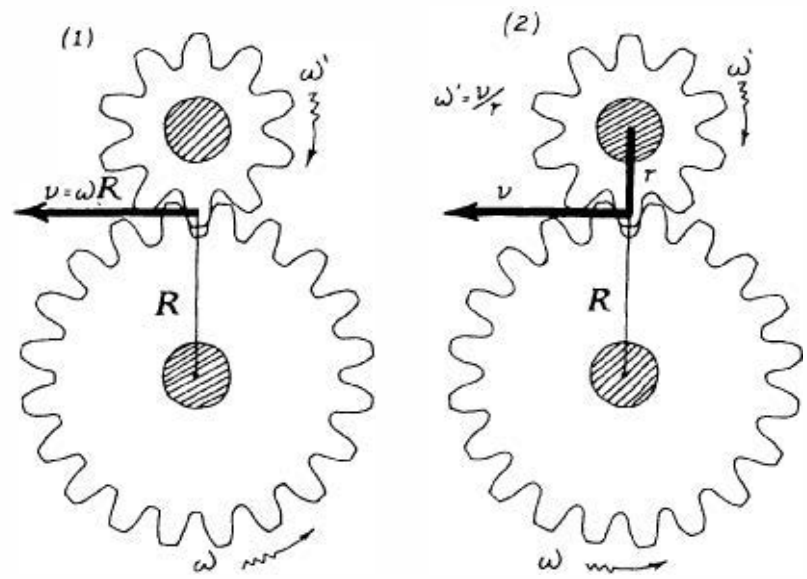


Fig. 1–20. Graphical construction for finding the angular velocity of one spur gear when the angular velocity of a mating gear is known.

(1) Known: ω, R.

(2) Construct v, as shown in (1), with a magnitude $v = \omega R$.

(3) Construct r as shown in (2).

(4) The angular velocity ω' of the small gear can now be found, since v is common to both gears. ω' equals v divided by r.

(rate-of-change of angular displacement as a function of time) are involved.

Consider, for example, the velocity of the pin on the disc shown in Fig. 1–17. When we first look at the pin (upper diagram), it has an instantaneous velocity v_t at right angles to the radius connecting pin and the shaft on which the disc turns, and is moving up and toward the right of the page. Let us assume the disc on which the pin is mounted has a constant angular velocity ω. A short time later, therefore, (lower diagram), the pin will have an instantaneous velocity v_t'. The length of velocity vector v_t' is the same as that of v_t, but the direction of motion of the pin has changed. If the angular velocity of the disc ω, had changed, then v_t' would have differed from v_t in both magnitude and direction. Note that the pin's linear velocity is always perpendicular to the radial line between pin and shaft.

There is a simple relationship between instantaneous linear velocity v_t or (v_t') and instantaneous

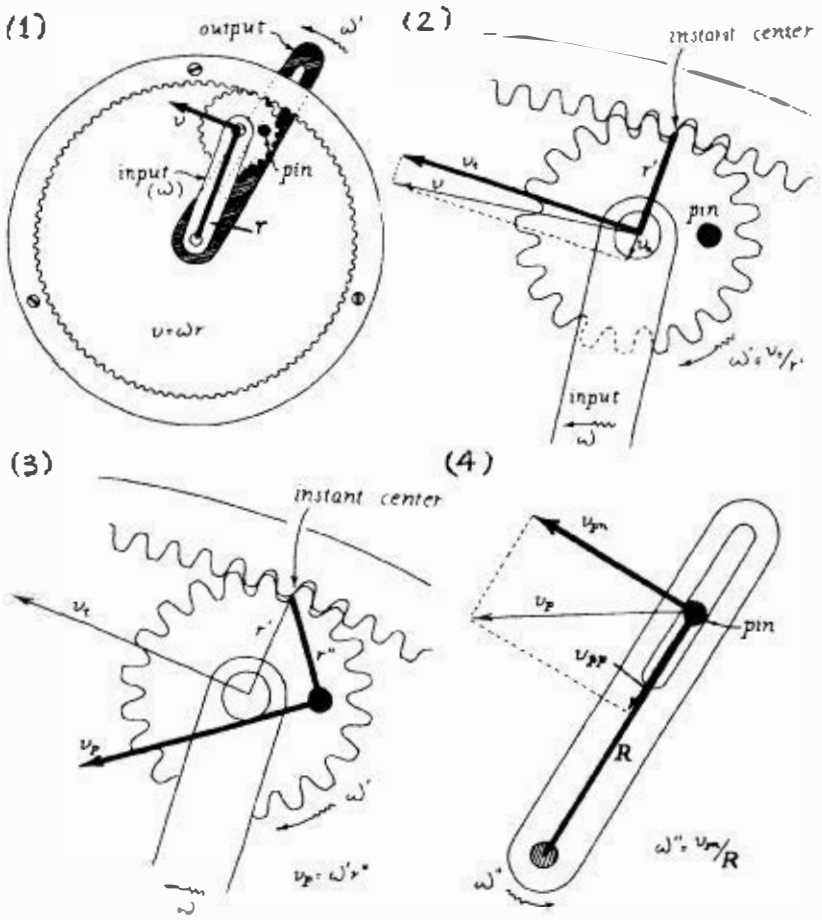


Fig. 1–21. (1) Hypocycloidal crank mechanism for which angular velocity ω' of the output crank is to be found graphically when input angular velocity ω and input crank radius r, are known. Instantaneous linear velocity v is constructed perpendicular to r with $v = r\omega$.

(2) Center of small gear is turning about instant center on radius r'. Construct instantaneous linear velocity v_t at right angles to r' from center of small gear. v_t is a component of v and hence its length can be found graphically as shown. (v_a is the other component of v at right angles to v_t and in line with r'.)

(3) Here the angular position of the pin is taken to be such that its radius r'' about the instant center is equal to r'. The instantaneous linear velocity v_p of the pin is now drawn at right angles to r'' and will be equal to v_t.

(4) Here the radius of the pin from the center of rotation of the output is shown as R. With instantaneous linear velocity v_p known, its component velocity v_{pn} at right angles to R can be found graphically as shown. (v_{pp} is the other component at right angles to v_{pn} and along radius R). The angular velocity ω'' of the output crank can now be found by the formula $\omega'' = v_{pn}/R$.

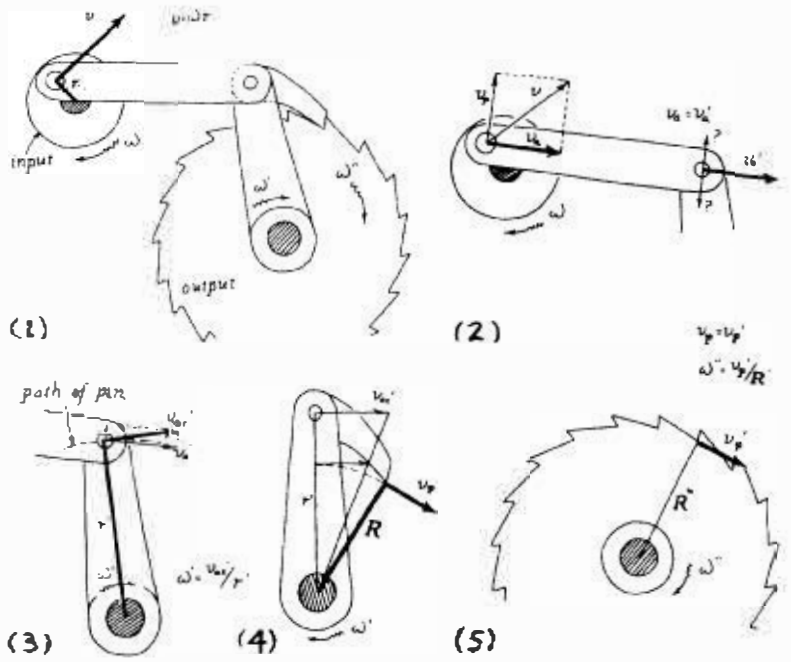


Fig. 1–22. (1) Ratchet wheel driven by crank with angular velocity ω and radius r of the input known. Instantaneous linear velocity v of the input pin is found by the formula $v = r\omega$.

(2) The components of v, one parallel to the axis of the floating link, v_a, and one at right angles to this axis, v_p, are then found graphically as shown. Instantaneous velocity, v_a', is the same as v_a since the two ends of the link cannot have different linear velocities along the axis without the link being either crushed or torn apart. Hence v_a' is constructed as shown. Note that the right-hand end of the link could be moving at right angles (and only at right angles!) to v_a as shown by the vectors bearing question marks.

(3) With v_a' known, the instantaneous linear velocity v_{at}' of the end of the next (vertical) link at right angles to its axis can be found graphically as shown. (Note that any other motion of this end of the link must be at right angles to v_a', in this case, upward.) With linear velocity v_{at}' and radius of rotation r' of the vertical link known, the angular velocity ω' of this link can be found from the equation $\omega' = v_{at}'/r'$.

(4) The vertical link and the ratchet pawl together can be assumed to be a single free body for the instant at which we are determining the linear velocities of the various members of this mechanism *train*. With v_{at}' known, the construction methods used in Fig. 1–19 can be used to find the velocity v_w of the tip of the drive pawl.

(5) Since the pawl engages the ratchet wheel they must have a common instantaneous linear velocity at the point of contact. Thus, v_w' is shown equal in length to v_w and at right angles to R', its *radius* to the center of rotation of the ratchet wheel. The angular velocity of the ratchet wheel ω'' can now be found from the equation $\omega'' = v_w'/R'$.

There is a simple relationship between instantaneous linear velocity v_t or (v_t') and instantaneous

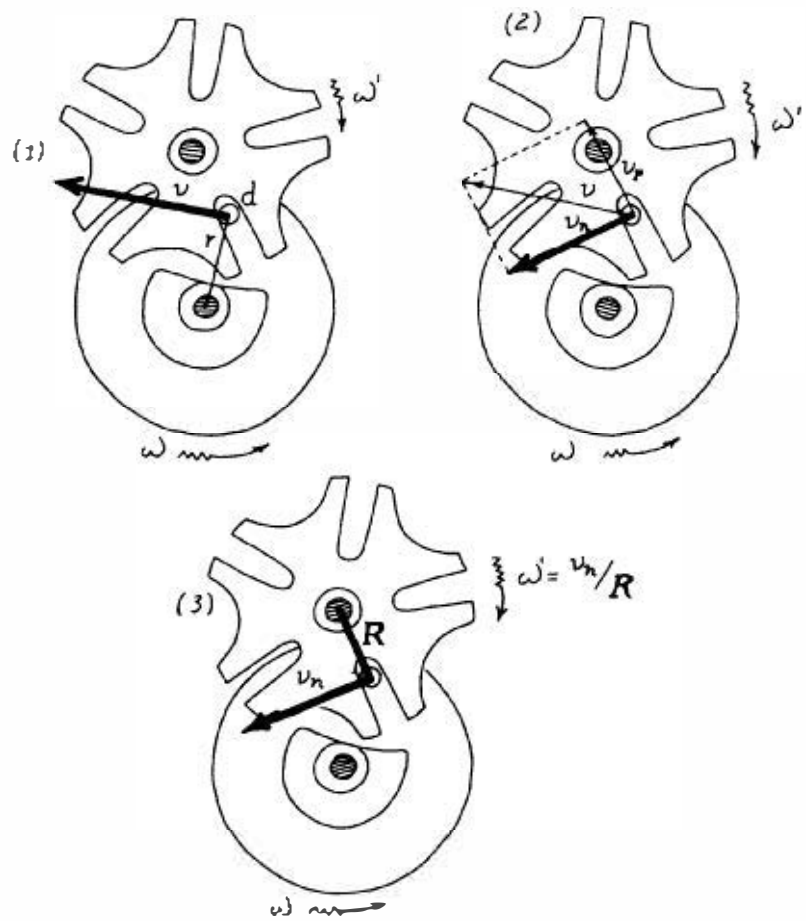


Fig. 1–23. (1) Geneva wheel for which instantaneous angular velocity ω' is to be found when angular velocity ω of input wheel and radius to drive pin r are known. This example illustrates the method of finding the velocity of a driven member when the drive pin moves in a slot in the driven member. Instantaneous velocity v of the drive pin d at right angles to r is found from the equation $v = \omega r$.

(2) Instantaneous linear velocity v is now divided into two components, one parallel to the axis of the Geneva wheel slot, v_p, and one at right angles to this axis, v_n.

(3) With v_n and radius R known, ω' can be found by the equation: $\omega' = v_n/R$. It should be noted that the selection of the components of velocity v was determined by the desire to find ω' using the formula $\omega' = v_n/R$.

angular velocity ω; it is:

$$v_t = \omega r$$

$$\text{Units: } \frac{\text{ft}}{\text{sec}} = \frac{\text{radians}}{\text{sec}} \times \text{ft}$$

where r is the radial distance between the pin and the center of rotation.

We see another illustration of this relationship in Fig. 1–18-(1) where we have a crank with two pins. To find the velocity of the inner pin, we first measure the radius (r), between that pin and the center of rotation, p. The instantaneous tangential velocity of that pin is then shown by vector v, in Fig. 1–18-(2)a Vector v is perpendicular to radius r (or tangent to the arc described by the pin as the link turns), and is equal in length to the product of r and ω, the latter being the angular velocity of the link.

The instantaneous velocity of the outer pin can be found by multiplying its radial distance from p, by ω, or by constructing the triangle shown in Fig. 1–18-(3). Notice that large V and small v are parallel to each other, and that their magnitudes are proportional to their relative distances from the center of rotation to the link.

Other Constructions for Determining Machine Velocities

The triangle construction technique used to find V in Fig. 1–18 can help us visualize and determine velocities of machine members in much more complex design situations than that represented by the single link of Fig. 1–18. We will usually use algebra to get numerical values for the forces, velocities, etc., of interest in design situations, but should use graphics or sketches to visualize what is happening and, therefore, to determine the correct algebraic equations to use. Figures 1–19 through 1–23 show graphical determinations of velocities for irregular links, a pair of simple gears, an hypocycloidal gear-crank mechanism, a fourbar ratchet-drive mechanism, and a Geneva mechanism, respectively. If you can understand each of these constructions you should be able to determine the velocities of any mechanism described in this book.

A Word About Units

A detailed discussion of units occurs later, in Chapter 2. In the illustrations of this first chapter we can use:

torque (τ) ft-lbs
force (F) lbs
radius (R, r) ft
linear velocity (v), ft/sec
angular velocity (ω), radians/sec.

CHAPTER 2

Graphical Mechanics and Intermittent Motion

Several concepts in basic mechanics were reviewed in Chapter 1, as well as methods of handling force and velocity vectors when analyzing mechanism designs. But all this involved only instantaneous velocities or instantaneous forces. Most design situations are concerned with a continuous spectrum of forces and velocities. If we had to compute these things separately for each instant of a long and complex machine cycle using the techniques of Chapter 1, alone, it would soon prove to be too difficult. Fortunately, in most design situations, approximate "motion curves" of force and velocity can be constructed, showing the complete and continuous variation of these parameters, and the need to find exact instantaneous values of force and velocity will exist only for a few special points on the curves (maximums and minimums, for example). Frequently, these approximate curves will be constructed using design information; e.g., when a spring is to be supplied capable of producing a certain force to drive given mechanisms. Some curves will result from experimental information, as described in Chapter 6, where a transducer, fastened to the machine, produces electrical signals that are "plotted" by a strip chart recorder or oscilloscope. And finally, there are curves derived from others, in some situations. In this chapter we will examine these motion curves, and see how they relate to each other.

This will also lead into a discussion of acceleration and displacement. These parameters will be handled as scalar quantities rather than vector quantities (which is why they were not discussed in Chapter 1), since only their magnitudes are of interest here. In conclusion, the annoying, but essential, subject of "units" will be discussed.

Forces on Mechanisms

Every element in a machine or instrument is acted upon by forces produced either by springs; electrical, pneumatic, or hydraulic motors; gravitational or magnetic fields; or by interactions with other machine elements and structures. In general, such forces can be considered to be either driving forces or load forces, and can be expressed as functions of time or functions of displacement. The discussion will be restricted almost exclusively to forces which are described as functions of time, but will also briefly cover the case of force versus displacement.

Forces, as have been shown, are vector quantities and are, therefore, directional in nature. It is useful to adopt "plus- or minus-sign rules" to describe force vectors so that our theoretical models can account for the fact that some forces act together while others act to oppose each other. For the purpose of this text, forces will be considered to be *positive* if they are directed toward the right, or vertically upwards. They will be considered to be *negative* if they are directed towards the left, or vertically downwards. These and other sign and symbol conventions used in this text are illustrated in Appendix 1.

Force and Intermittent Motion

Let us start our study of force with a very simple and tangible case of intermittent motion. In Fig. 2–1, a finger applies a brief push to a small block resting on a table. This produces a drive force F, on the stone and inevitably, a friction reaction force f_r. After the finger has stopped pushing (comes to rest), the block coasts for a while until the frictional force brings it to a rest. This action, plus the resulting graph of net horizontal force (drive force minus friction force) on the block, is shown in Fig. 2–2, A and B.

It would be quite easy to measure the drive force involved here (with a push scale for example). Measuring the friction force would be a slightly more difficult problem, but it can be done by tipping the table to determine the sliding angle, for example, and a graph can be produced of the force as a function of time, such as is shown in Fig. 2–2B.

This would immediately disclose one important fact: the positive and negative areas enclosed by the force curve (and time axis) must be equal to each other if the velocity of the block is to be zero at the beginning and end of the period being studied. In other words, if the block is to be a true "intermittent motion device" it must start with zero velocity and end there. This equality of the positive and negative areas enclosed by a force (or torque) curve is an important and basic concept in the study of intermittent motion.

Notice that in the graph, Fig. 2–2B, force is expressed in pounds, and time in seconds. These are two quantities of the so-called English system of units which will be used throughout this text and which is tabulated in the appendix. Also given in the appendix, however, is a table of alternate sets of units involving ounces, grams, inches, kilograms, etc. Be consistent and stay within one set of units (one column in the table) for a given problem; it is only when the sets of units are intermixed that you will get into difficulty.

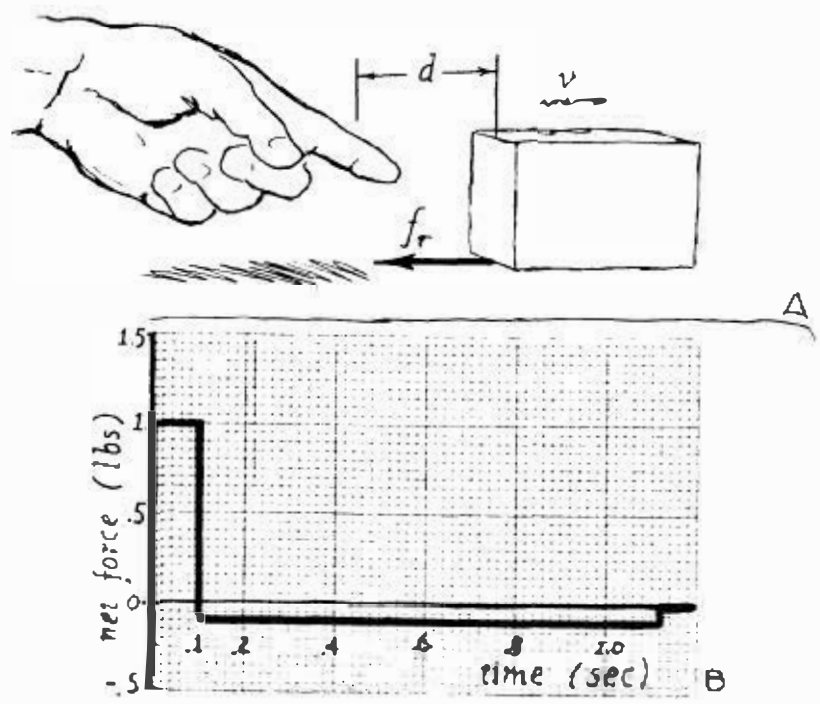


Fig. 2–2. (A) Forces acting on a sliding block; and (B) Graph showing the net or resultant horizontal force on the block as a function of time.

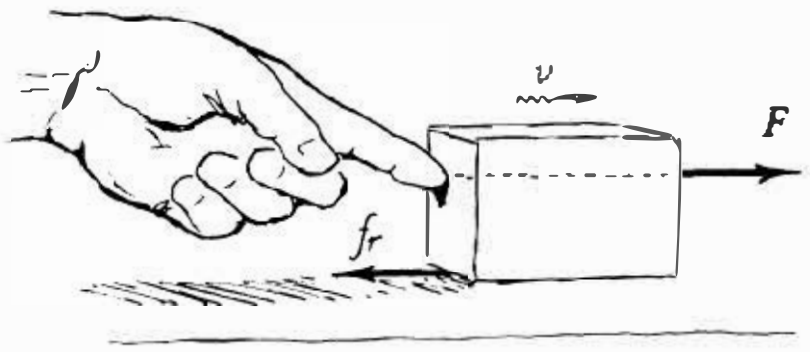


Fig. 2–1. Forces acting on a block being pushed along a flat surface.

Consider now, another topic, the acceleration of the body and its relationship to the forces on that body.

Acceleration

Whenever an object changes its rate of motion; whenever it speeds up or slows down, it experiences positive acceleration (an *increase* in speed), or negative acceleration (a *decrease* in speed). The latter, of course, is sometimes called deceleration but "negative acceleration" is the more accurate term for theoretical model builders. Acceleration is defined as the rate-of-change of velocity as a function of time. It is a very important parameter in machine design and is, therefore, worthy of our attention. Fortunately, once the forces on a body are known, its acceleration can readily be determined. The acceleration (as a function of time) is found by using Newton's famous Second Law:

$$F = ma$$

force = mass × acceleration

(units: lbs = slugs × ft/sec/sec = slugs × ft/sec^2)

This law tells us that the acceleration of a body at any instant is equal to the force on the body at that instant divided by the inertial mass of the body. A force curve, then, can be converted to an acceleration curve merely by dividing each point on the force

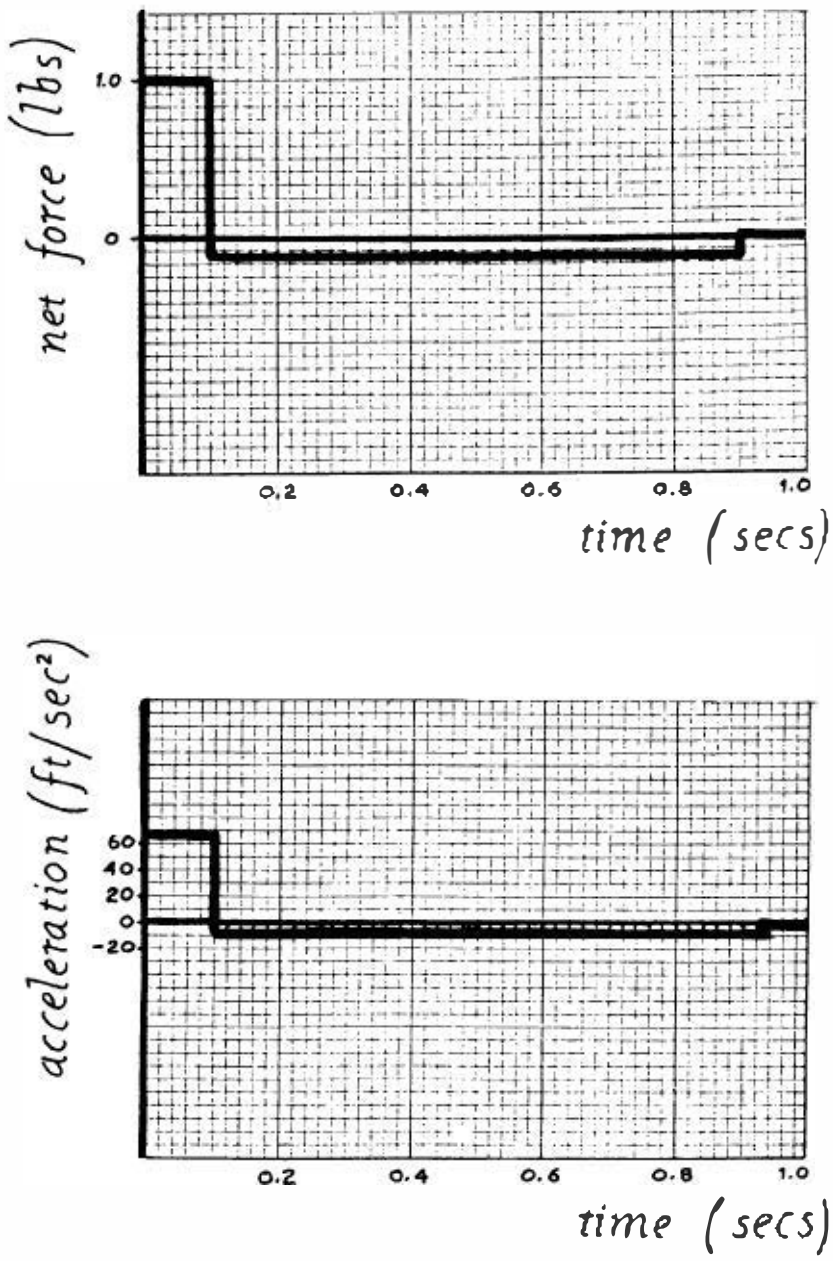


Fig. 2–3. Relationship between acceleration and force curves for the block-on-the-table problem.

curve by the mass of the body being studied. This can be done by drawing a new acceleration curve using information obtained from the force curve; or by merely changing the vertical scale on the original force curve. Figure 2–3 shows the relationship between the forces and accelerations on the block. Note that the positive and negative areas enclosed by the acceleration curve are equal for intermittent motion.

The units for acceleration are feet per second per second, which equals feet per second, squared; again, in the English system of units. Time is still given in seconds. Mass has been expressed in slugs. I do not know the origin of the name "slugs" nor why the word was adopted. By definition (and using Newton's equation of $F = ma$), a unit force of 1 pound will produce an acceleration of 1 foot per second, squared, in a 1-slug mass: thus a slug is related to more familiar units by the following equation:

$$M = f/a$$

$$\text{One slug} = \text{one } \frac{\text{lb-sec}^2}{\text{ft}}$$

—if that is any help! We can find the mass of a body by dividing its sea-level weight, in pounds, by 32. (32 ft/sec² being the acceleration of gravity in English units. 32.16 or 32.2 are sometimes used but 32 is close enough for our purposes.)

Once an acceleration curve is achieved, a velocity curve can be constructed. Velocity is also of interest in design work, and is the next topic to be discussed.

Velocity

Velocity is defined as the rate-of-change of position of a body with time. It is a measure of the speed at which a body is moving; and the direction in which it is moving. Because velocity, like force, is a vector, positive and negative sign conventions must be adopted. In this text we will assume that velocity to the right, or upward, is positive: and velocity to the left, or downward, is negative.

If the acceleration of a body as a function of time is known, a velocity curve can be constructed by measuring the area enclosed by the acceleration curve and the horizontal (time) axis on the acceleration diagram. This area could be found by a mathematical process called *integration* if the acceleration curve could be described mathematically. There are many special cases in which this is possible, and some students struggle through a course in integral calculus to learn them. With a drawing of an acceleration curve, however, the integration process can be performed graphically, merely by counting the squares enclosed by the curve and the time axis, by using an instrument called the planimeter, or by a series of algebraic computations in which the area under a curve is approximated by a series of rectangles, triangles, trapezoids, and other simple forms whose areas are easy to compute. The same techniques can be used for finding the areas under complex curves as for finding those areas under simple curves. It is never necessary to learn special procedures for special cases; an advantage of the graphical over the mathematical approach.

The integration process is started at the "velocity equals 0" point. This is not an arbitrary choice, it is necessary to start at the zero velocity point to avoid what mathematicians call the "constant of

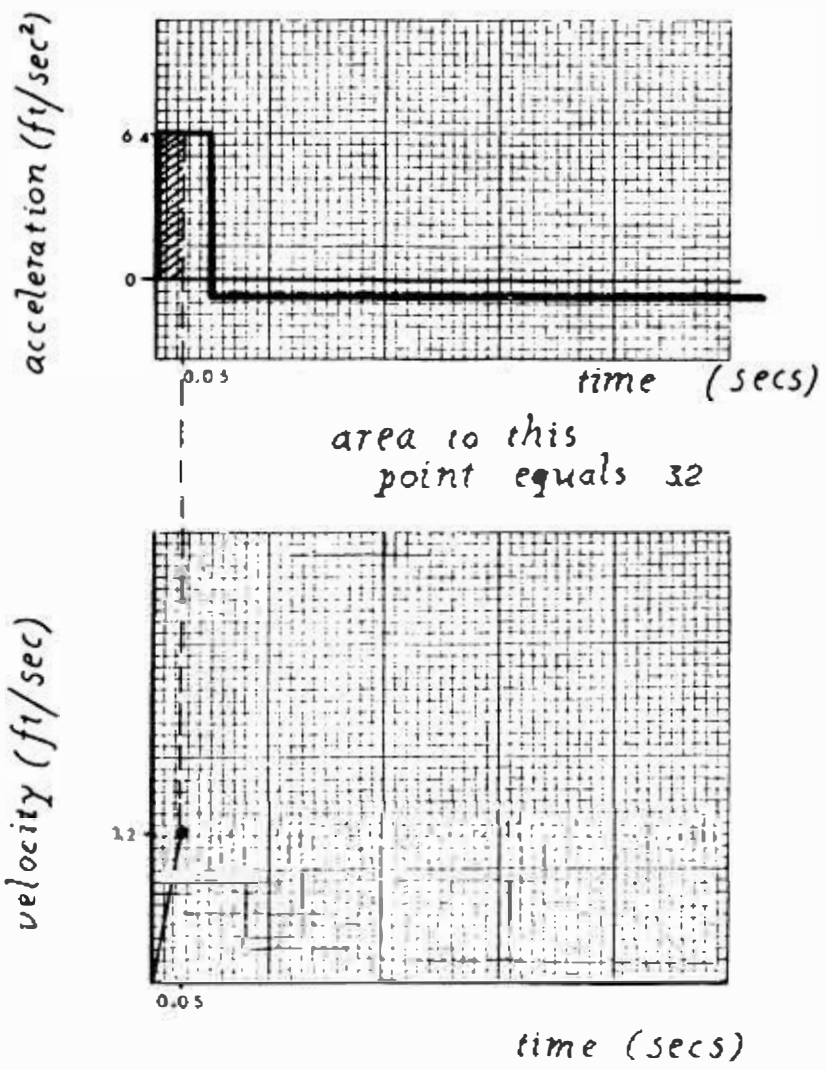


Fig. 2-4. Deriving a velocity curve from an acceleration curve—first step.

integration." The velocity of the body after 0.05 second of acceleration is equal to the total area enclosed under the acceleration curve from the time (or velocity) equals zero point to the time equals 0.05 second point. The velocity of the body after 0.1 second of acceleration is equal to the total area under the acceleration curve from time = 0, to time = 0.1 second, etc., as shown in Figs. 2-4 and 2-5.

When the acceleration curve passes below the time axis it produces the "negative area" which must be subtracted from the total area accumulated to that point, as shown in Fig. 2-6. In an intermittent motion system, the velocity curve starts and stops at zero; thus, as shown in Fig. 2-7, as much area is eventually subtracted as was added. This process of graphical integration is very important as the methods of this text cannot be used unless the process is thoroughly understood. Re-read the above and study the illustrations until the process of converting the "force-on-the-block" curve into a velocity curve is clearly understood.

The integration process can also be used to shed light on the "units" that should be used for the function being derived. Acceleration, as was seen, has the units of feet per second, squared. Time is measured in seconds. The area enclosed by an acceleration curve and the time axis, therefore, would have the dimensions of:

$$\text{Area} = \text{height} \times \text{base}$$

$$\text{Units: area} = \frac{\text{ft}}{\text{sec}^2} \times \text{sec}$$

It is a very useful, but little-known fact that "units" can be treated as algebraic quantities in trying to simplify an expression or to see where an equation is heading. For example, just as:

$$\frac{A}{B^2} \times B = \frac{A}{B}$$

so does the original "units" expression given above,

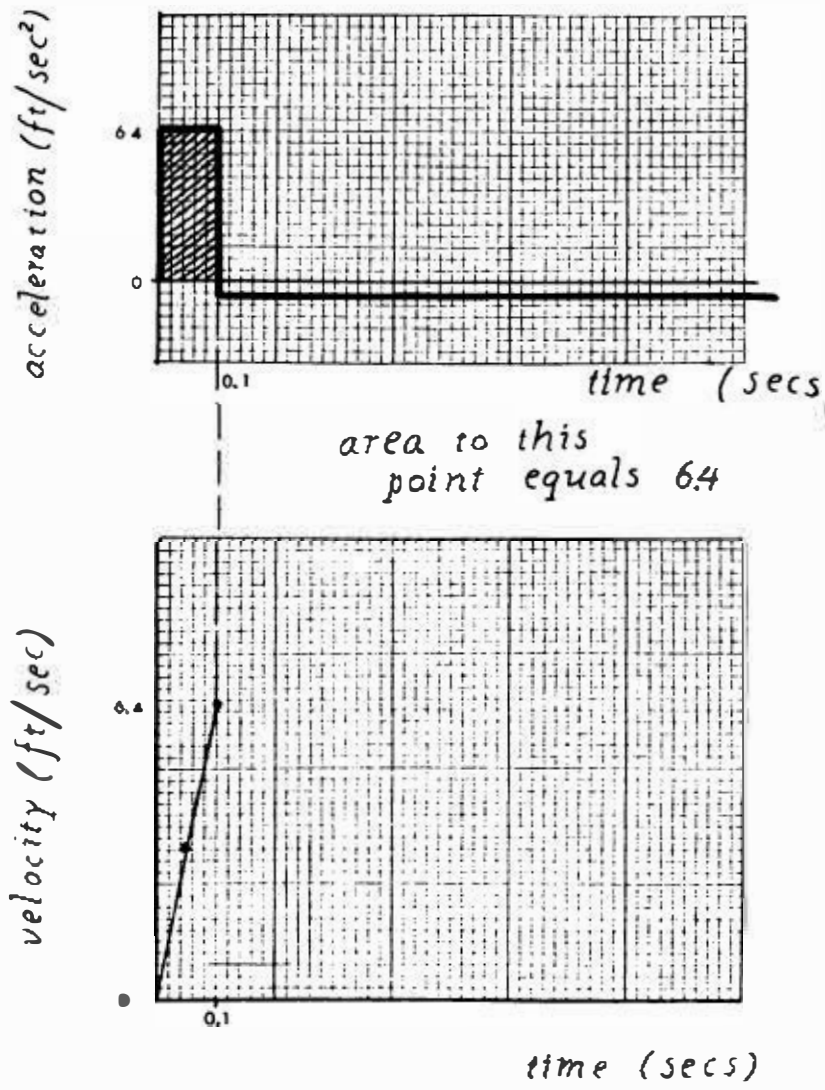


Fig. 2-5. Deriving a velocity curve from an acceleration curve—second step.

reduce to feet per second, as shown below:

$$\frac{\text{ft}}{\text{sec}^2} \times \text{secr} = \frac{\text{ft}}{\text{sec}}$$

And feet per second is indeed the correct "unit" for linear velocity. Therefore, although it may seem strange, the "area under the acceleration curve" really *is* "velocity."

Units as Algebraic Quantities

Again, this treatment of units should be reviewed before leaving this section. Nothing causes the beginner more grief than units, and this method of handling them will be helpful. I believe it comes from that branch of mathematics called dimensional analysis.

This method works with any group of units. For example, suppose there is a truck that holds 500 pies; each pie takes up 200 cubic inches; has a density of 100 ounces per cubic foot; and it is necessary to know the total weight of the 500 pies in

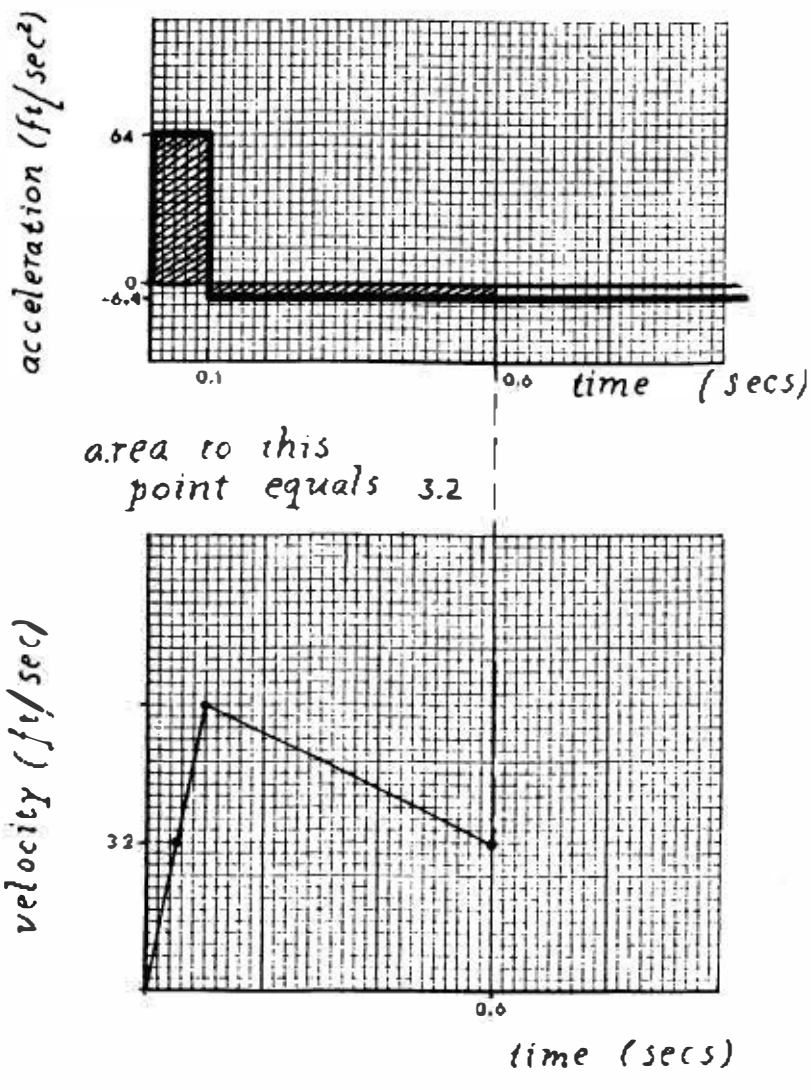


Fig. 2–6. Deriving a velocity curve from an acceleration curve—third step.

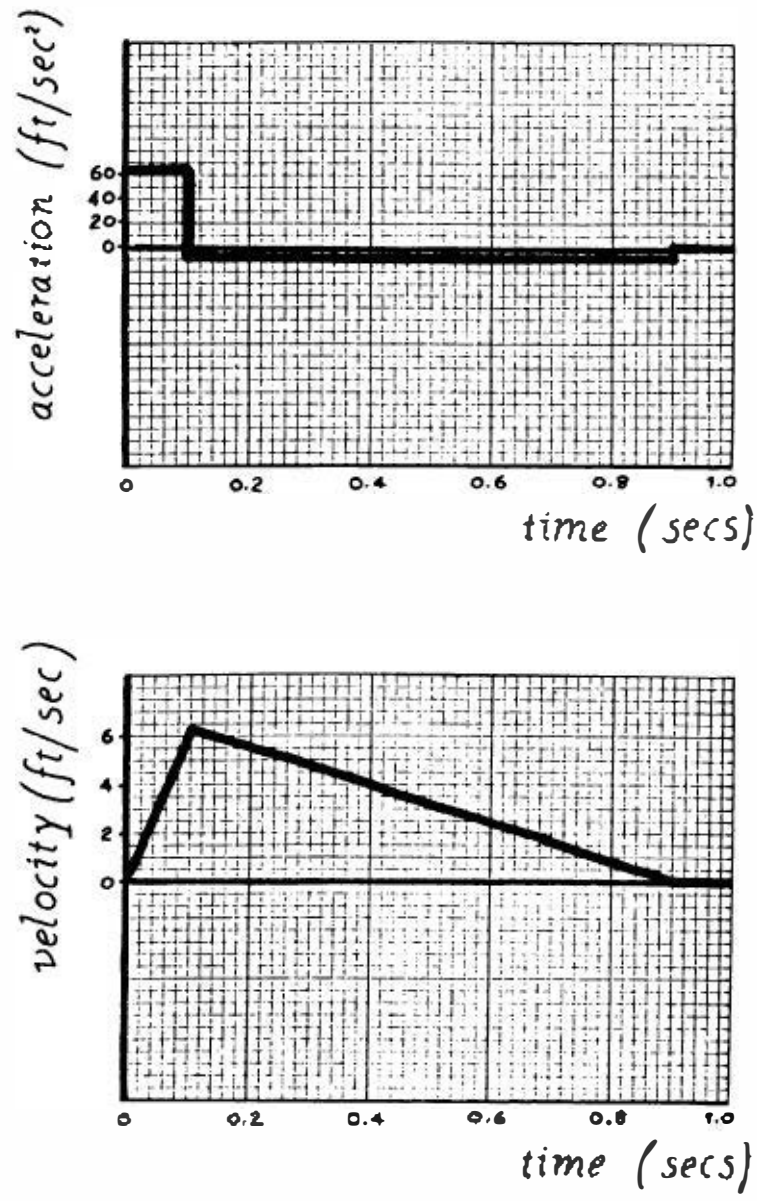


Fig. 2–7. Completed relationship between acceleration and velocity curves for the block-on-the-table.

pounds! If the three known "things" are multiplied together and algebra is used to "cancel" the "pies" unit the result is:

$$500 \text{ pies} \times 200 \frac{\text{in}^3}{\text{pies}} \times 100 \frac{\text{ounces}}{\text{ft}^3}$$

$$= 10{,}000{,}000 \frac{\text{in}^3 - \text{ounces}}{\text{ft}^3}$$

—which sounds meaningless. Somehow we must get rid of the in^3 and ft^3 terms and convert OUNCES to POUNDS. To do this multiply the original expression by conversion factors:

We know that $1\ \text{ft}^3 = 1728\ \text{in}^3$, or that

$$\frac{1\ \text{ft}^3}{1728\ \text{in}^3} = 1$$

We also know that 16 oz = 1 lb, or $\frac{1\text{ lb}}{16\text{ oz}} = 1$

The "pie" equation can now be multiplied by these two expressions for "1" to change its units without changing its true value:

$$500 \text{ pies} \times 200 \frac{\text{in}^3}{\text{pie}} \times 100 \frac{\text{oz}}{\text{ft}^3}$$

$$\times \frac{1\text{ ft}^3}{1728\text{ in}^3} \times \frac{1\text{ lb}}{16\text{ oz}} = 362 \text{ lbs}$$

or 0.72 pound per pie.
All units have cancelled algebraically except the one wanted, pounds, so there is the answer. Practice this method of checking units. It is extremely worthwhile.

Displacement

Getting back to graphical mechanics: once there is a velocity-versus-time curve we can graphically integrate again—finding the area under the velocity curve this time—to get a curve of the displacement of the block on the table as a function of time. Use exactly the same graphical procedure as in going from the acceleration curve to the velocity curve, first finding the area between "time = zero," and "time = 0.1 sec," and setting this equal to the displacement at 0.1 sec, etc.

Again, the units of the area of the original curve (in this case, velocity) become the units for the vertical axis of the derived curve (in this case, the displacement curve), as can be seen by the following equation:

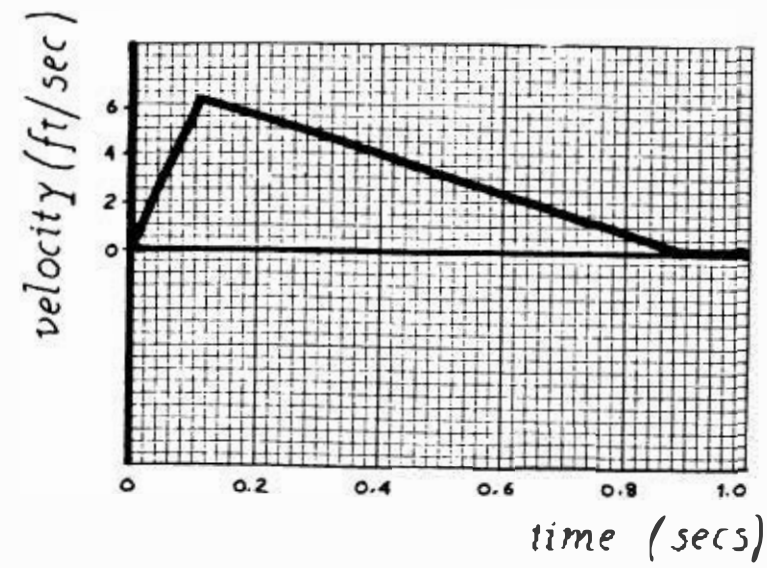
Area under the velocity curve = height × base

$$\left(\text{area} = \frac{\text{ft}}{\text{sec}} \times \text{sec} = \text{ft}\right)$$

and feet, of course, is the correct English unit for displacement.

Figure 2–8 shows the velocity curve of Fig. 2–7, repeated, and the displacement curve that is derived from it. Again, if graphical integration is still not understood, please re-read the discussion. It is essential that you understand it if you are to use the methods of this text.

Note that the displacement curve does not return to zero in this case. Although the block has started from rest, moved, and stopped again, it retains its new displacement, or position. This is typical of many intermittent motion devices.



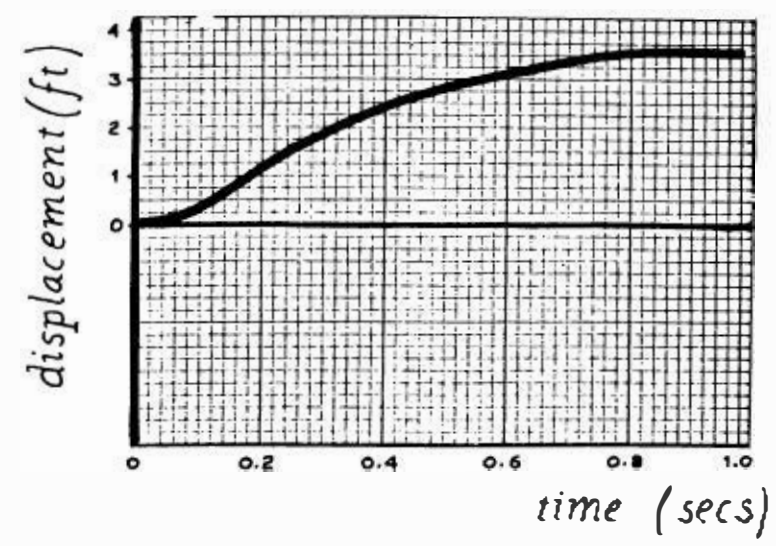


Fig. 2-8. Relationship between velocity and displacement curves for the block-on-the-table.

Exact Numerical Values by Graphical Integration

The process of graphical integration has now led from a force curve (acceleration curve after the vertical scale was changed), to a velocity, and then a displacement curve. This provides all of the necessary information about a mechanism, to analyze its performance. However, the techniques of Chapter 1 must be used to find critical values on the curves of force and velocity if exact numerical values for the various parameters are wanted, because the graphical integration process is a method for obtaining approximate results only. It is used for insight, and to reduce the number of points for which exact numbers are calculated, rather than for definite numerical data.

To get exact accelerations the methods of Chapter

1 must be applied to find instantaneous forces, then Newton's Second Law is used to convert these to accelerations. Or the methods of Chapter 1 can be used to find exact values for velocity just before and just after the point at which it is desired to determine the acceleration; and then use the equation:

$$\text{average acceleration} = \frac{v_2 - v_1}{t_2 - t_1}$$

to determine the average acceleration between times t_1 and t_2. If t_1 and t_2 are nearly equal, the average acceleration will equal the instantaneous acceleration at that point.

There is no really good way to find exact numerical values for displacement by graphical integration. Here it is better to study the geometry of the parts involved and to attempt to relate displacement to the motion of the input link or shaft of the machine. If this input is running at a constant velocity (and only then!), it can be used as a "time base" and output displacement plotted versus input displacement, as shown in Fig. 2–9. Input displacement can be used as a base for all curves; displacement, velocity, and acceleration; and graphical integration can still be used to get the *shape* of lower curves. Cam designers frequently use displacement of driver for the horizontal axis. But be sure input displacement is converted to time before equating area under a curve to height of the next curve, in numerical terms. Ft/sec times secs = ft; but ft/sec times radians (input displacement) has no real meaning.

Deriving a Velocity Curve from a Displacement Curve

So far, we have seen how to derive acceleration, velocity, and displacement curves when starting with an experimental or predicted curve of force on a body as a function of time. It is often difficult or impossible, however, to determine what the force "picture" should be, so a procedure must be developed for going in the other direction; for starting, let us say, with a displacement-versus-time curve and from it deriving velocity, acceleration, and force curves. In a calculus course this process is called "differentiation." It can be done mathematically only if we can describe the original curve mathematically. For our purposes, however, it will be done graphically and without the mathematical description of the curve. As a result, extremely complex curves can be handled nearly as readily as simple curves. The procedure will always be the same, at any rate.

A curve, is differentiated by "measuring its slope" at a number of points. The slope at each point is found by drawing a line tangent to the curve at each point, and applying the equation: slope = H/B, as shown in Fig. 2–10. Note that the "slope" is equal to the tangent of the angle θ. Slopes, like areas, can be considered to be positive or negative in sign. They are *positive* if the tangent line is directed *upward* from left to right, as in Fig. 2–10; and are *negative* if the tangent line is directed *downward*, from left to right.

Notice that the process of calculating the slope can be used to determine the units for the quantity being derived; just as it was possible to use the process of taking areas under a curve to determine the units for the quantity being derived. In taking the derivative of a displacement-versusetime curve, for example, the vertical axis has the units of feet, and the horizontal axis has the units of seconds; treating these quantities algebraically, we determine

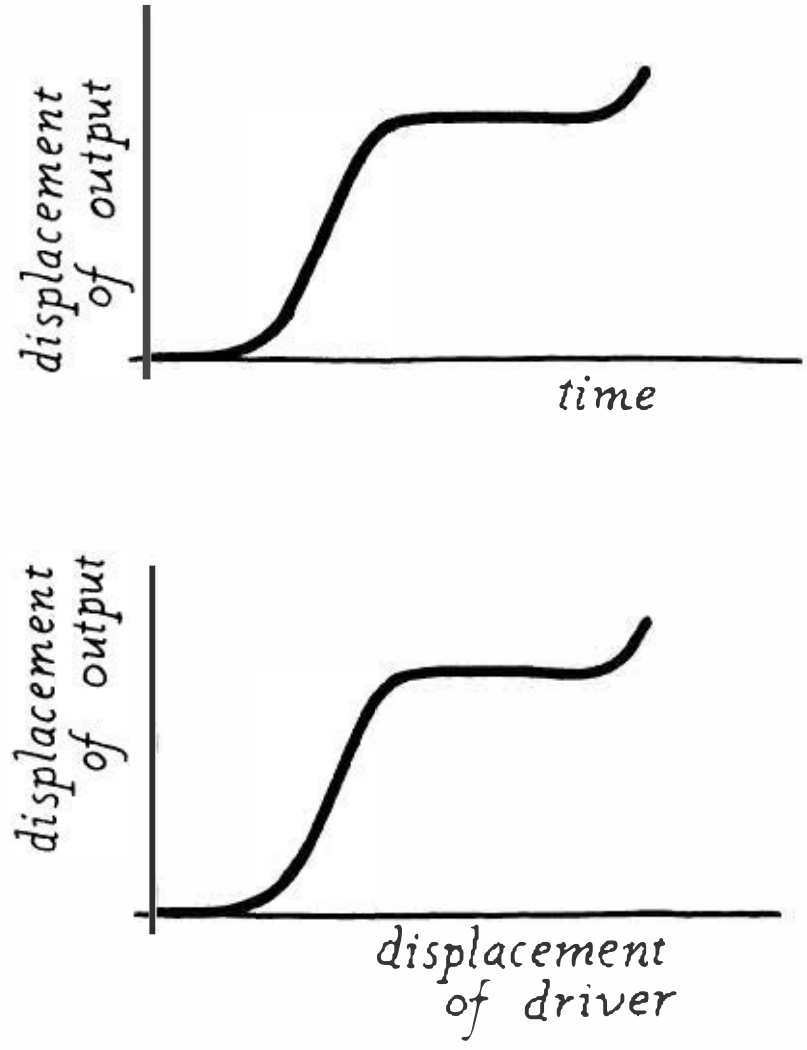


Fig. 2–9. Displacement of the "driver" can be used for a horizontal axis instead of time; but only if the driver is moving with a constant velocity.

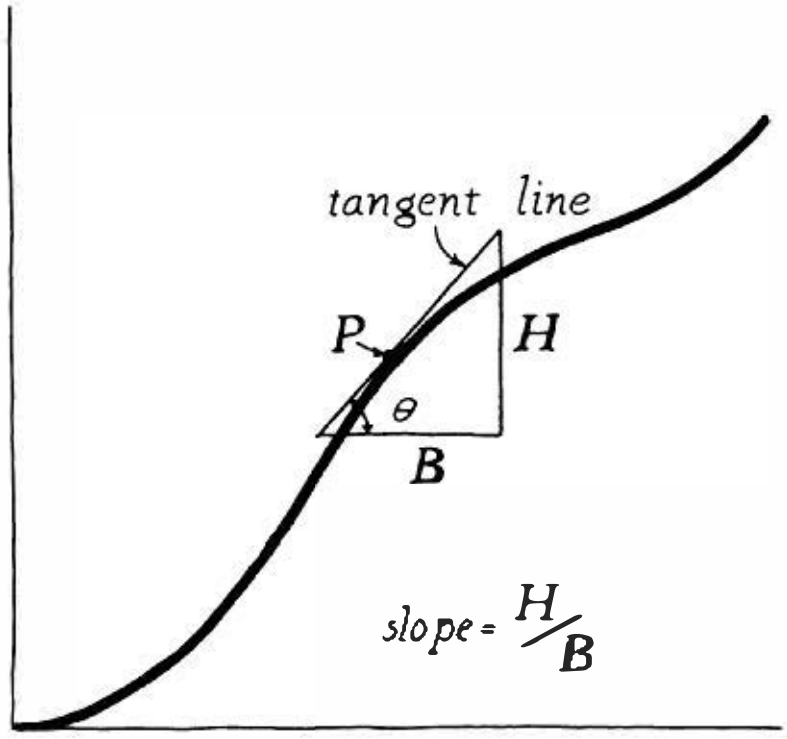


Fig. 2–10. Graphical differentiation—Determining the slope at point P.

that the units for velocity are feet per second, as shown below:

$$\text{slope} = H/B \text{ (from Fig. 2–10)}$$

$$\left(\text{units: slope} = \frac{\text{ft}}{\text{sec}}\right)$$

In Fig. 2–11 is seen the derivation of a complete velocity curve from a displacement curve. These graphs should be studied until their meaning is really apparent, since this process of taking derivatives is just as basic and important as is the process of taking areas. Notice that whenever the displacement curve is horizontal (parallel to the time axis), the derived or velocity curve is 0, for example. Notice that the maximum peaks on the velocity curve do not occur at times of maximum displacement, but at times of maximum rate-of-change of displacement. After some experience in graphical differentiation is gained, a derivative curve can often be roughed-in just by an inspection of the first curve, using a few special points such as those described above. This is discussed further, in a later section of this chapter.

In practice, slopes of curves can be taken by drawing tangent lines (by eye) on displacement curves and computing the slopes algebraically. There are instruments that can be used to draw accurate tangent lines in some conditions, and also other instruments for rapid calculation of the slopes of these tangent lines. Both are described in the appendix.

If one of these instruments is not available and, therefore, arithmetic or a slide rule must be used to compute numerical values for the slopes drawn, it will be a convenience to use the same base distance (B), for each little slope "triangle" drawn. This will simplify the many divisions needed; or the division steps can be skipped and the various triangles' altitudes (H values) can be plotted directly, to sketch the derived curve. Its shape will be correct as long as all B's are equal. Then the correct vertical scale can be computed and added, to complete the drawing.

Graphical differentiation, just as graphical integration, is not a precise process, and will not lend itself to exact answers except in special cases, but again, this need not be considered a serious problem. Exact answers for intermittent motion situations cannot be obtained anyway—usually, for example,

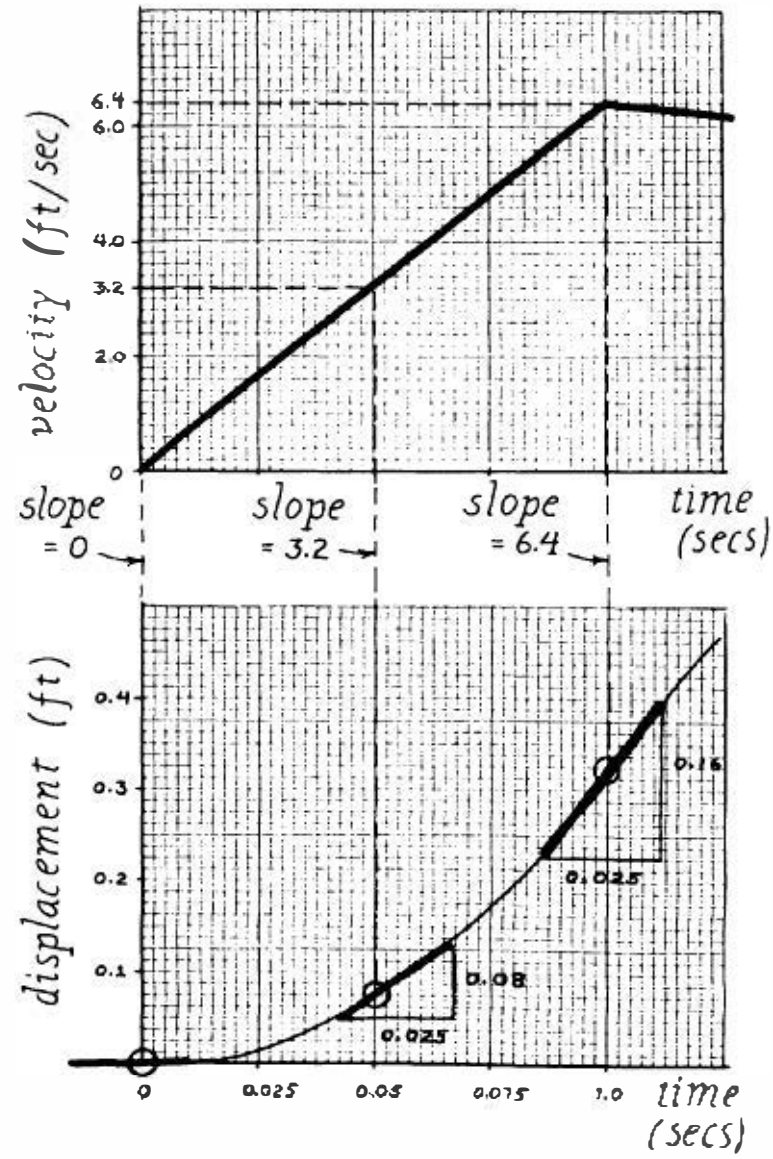
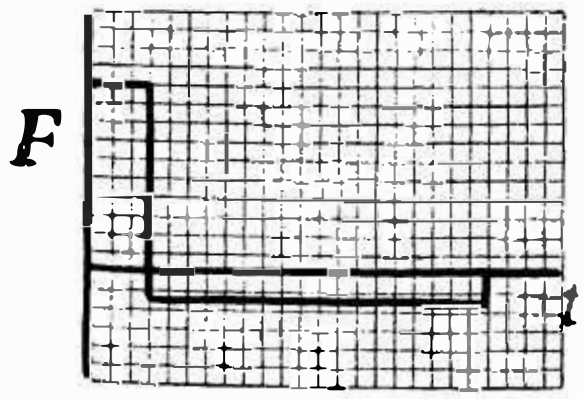
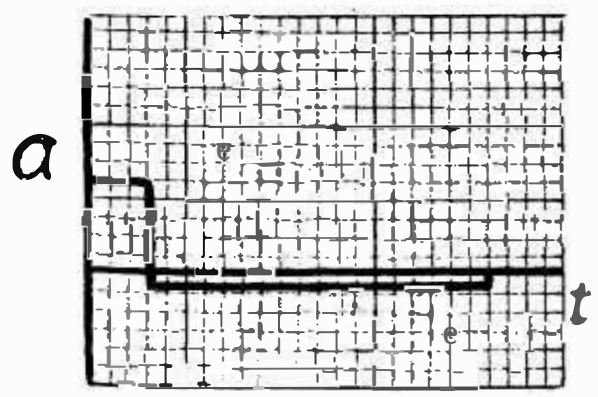


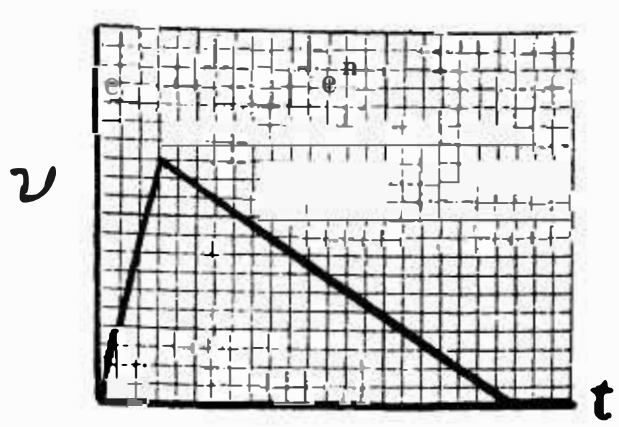
Fig. 2–11. Deriving a velocity curve from a displacement curve by graphical differentiation.



Divide F by "mass" m, to get a curve, below, or multiply a by m to get F curve, above.



Take areas under a to get v, below, or take slopes of v to get a, above.



Find areas under v curve to get d curve, below, or find slopes of d curve to get v curve, above.

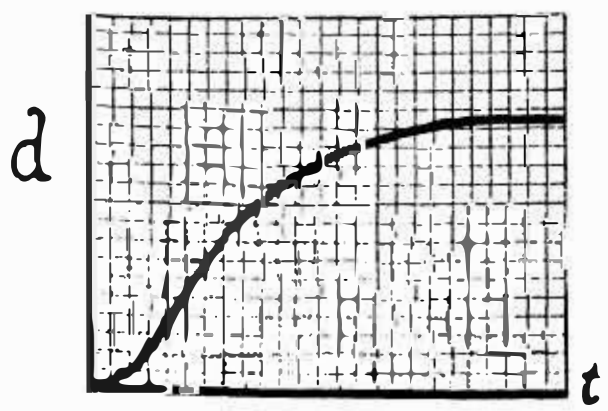


Fig. 2–12. The relationship between the force (F), linear acceleration (a), linear velocity (v), and linear displacement (d) curves.

it is impossible to write the equations of displacement as a function of the time which would allow exact mathematical solutions. The best that can be hoped for is to get an approximate feel for various situations, and this the graphical process does. Sometimes one can get "fairly exact" solutions for a few points on the curves, using the techniques of Chapter 1, for example, but even this is virtually impossible with many types of intermittent devices, as will be seen in later chapters. But the curves do permit a valuable insight to be obtained and they should be drawn.

From Velocity, to Acceleration, to Force

Using the process of graphical differentiation, an acceleration curve can also be plotted from the previously derived velocity curve. Then, using the familiar $F = ma$, each point of the acceleration curve can be multiplied by a constant factor (the mass of the body being studied) and a force-versus-time curve derived; reversing the process shown in Fig. 2–3.

Summary of Motion Curves—FAVD!

The text has now shown how to proceed from a force-versus-time curve to acceleration, to velocity, and then to displacement curves; and also how to proceed from a displacement-versus-time curve to velocity, acceleration, and force curves. It is now possible to enter the force-acceleration-velocity-displacement (FAVD) sequence at any point and to derive the rest. For example, if an experiment or design prediction gave a velocity curve as a function of time, graphical integration could be used to derive the displacement curve, and graphical differentiation to derive the acceleration curve. The acceleration curve would then be multiplied by mass to get a force curve. All that is needed in any problem is that first curve. Graphics will give approximate solutions for the rest. The vector methods of Chapter 1, or design data, will produce exact numerical values for critical points on these curves.

Figure 2–12 summarizes the relationships between the force, acceleration, velocity, and displacement curves for the block sliding on the table. Again, make certain that these relationships are understood before going further. This is not a difficult procedure once the sequence is apparent; but it is an essential development for the graphical understanding of intermittent motion devices. Perhaps the following will prove helpful in remembering the procedures:

Remember the relationships between the F-a-v-d curves and how to start with any one and derive the others.

"F A V D"—*F*ind *A*reas *V*en going *D*own.

"D V A F"—*D*ifferentiate *V*ery *A*ctively *F*riend, on going up!

But Be Careful!

Graphs, followed blindly, can cause as much trouble as a problem in mathematics followed blindly. Even though only one variable may be measured (force, or acceleration, or velocity, or displacement), there is probably some knowledge of the other factors in a given design situation. Make sure that your graphs appear to represent what you think the situation might indeed be.

Infinite Accelerations

Figure 2–13 shows a typical trap. It begins with a curve of displacement versus time and measurements

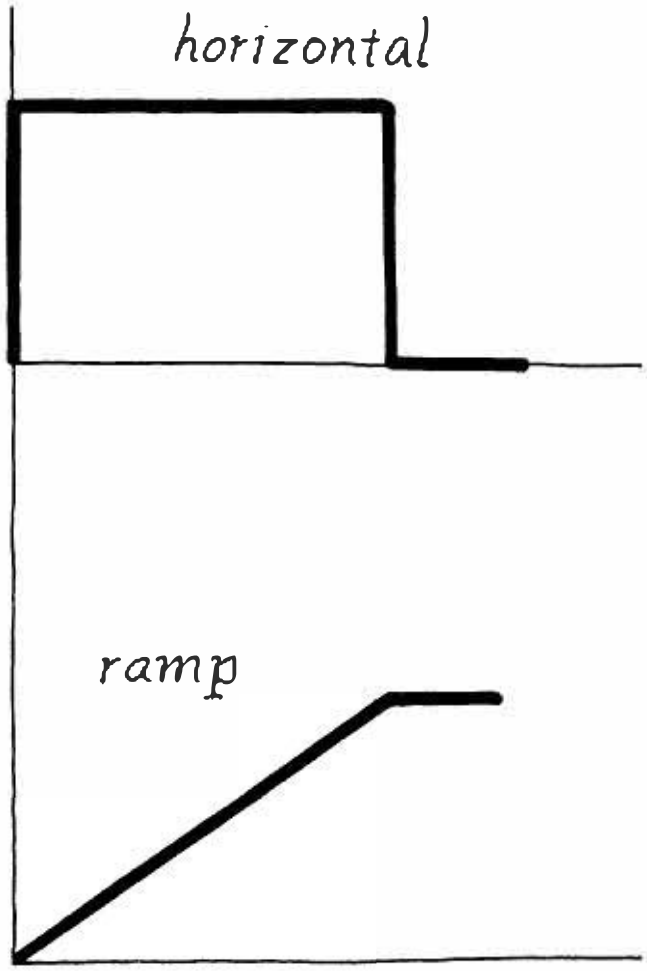


Fig. 2–13. Relationship between a ramp displacement curve and a step velocity curve.

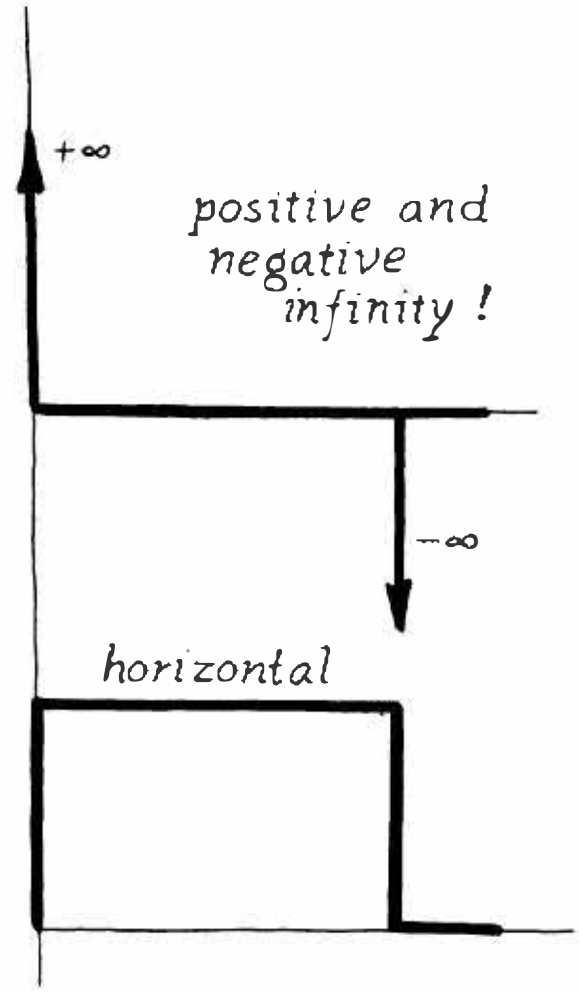


Fig. 2–14. A step velocity curve produces plus and minus infinite accelerations.

or design information indicating that this curve would be shaped like a ramp, as shown in the figure; with an upward slope from time equals zero to the inflection point of the curve. If this were the case, the velocity curve would consist of a straight line parallel to the time axis shown in the same figure. Notice that the velocity curve is vertical at its beginning and end: there was a finite velocity at the beginning and end of motion (just after, and just before, the body was at rest) says the displacement curve.

If we now go on to plot the acceleration curve (Fig. 2–14), remembering that acceleration is defined as the rate-of-change of velocity with time, we find that when the body starts into motion it must have an *infinite* positive acceleration. The velocity change was not very great (from zero to some finite level), but since it occurred in zero time (instantaneously) the *rate*-of-change, the acceleration, must be infinite.

$$\frac{\text{Change in velocity}}{0} = \infty$$

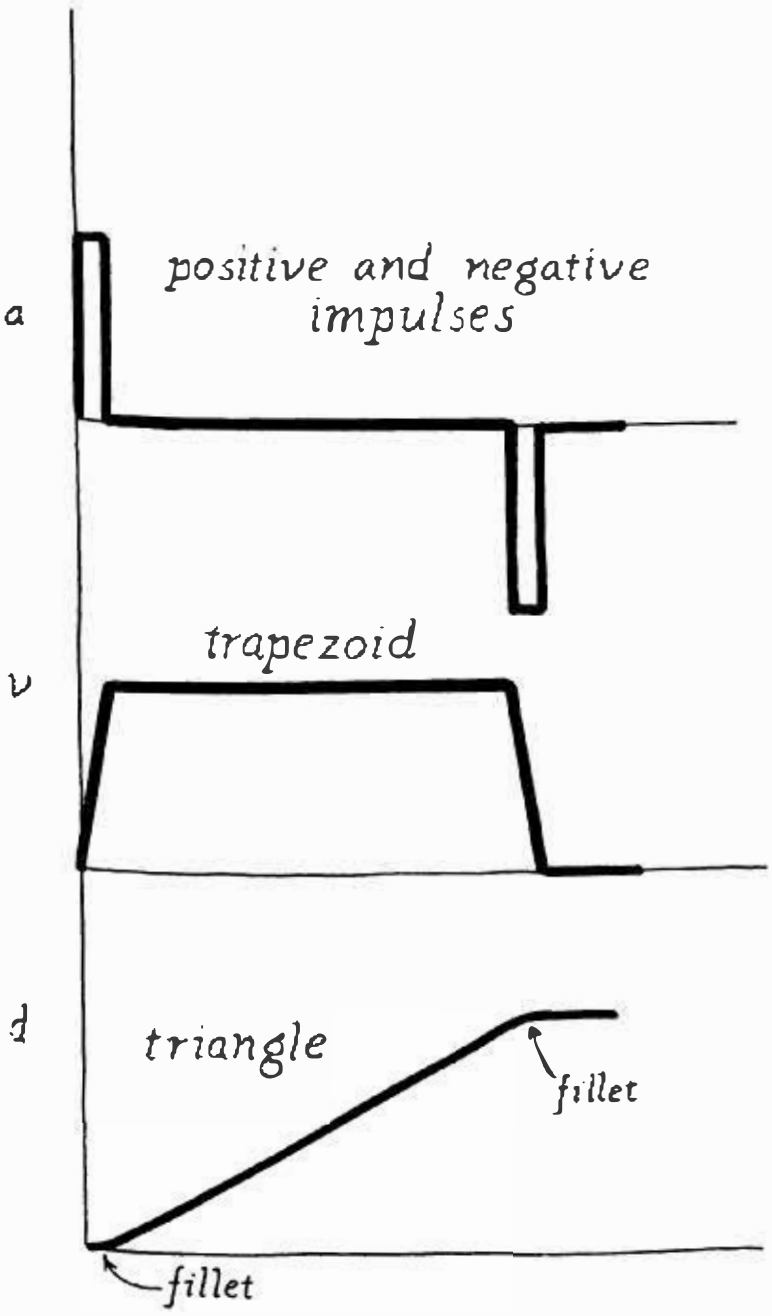


Fig. 2–15. Small fillets at the beginning and end of a ramp displacement curve produce a velocity trapezoid and finite positive and negative accelerations.

By the same token, an infinite negative acceleration must later be applied, to bring the body to rest. Since we obtain force curves by multiplying the acceleration curve by the mass of the body, it is apparent that an infinite force would have to be applied to the body in order to accomplish this apparent displacement as a function of time.

Of course, infinite forces and accelerations are not possible. What actually happens in a machine design situation is that the basically triangular displacement-versus-time curve will have small "fillets" at the beginning and end, as shown in Fig. 2–15. There would be, in other words, a finite period of time during which the velocity changed from zero to the essentially constant positive value, and a second finite period of time during which the body was brought to rest. Very frequently, these tiny fillets will be virtually impossible to measure, especially with intermittent motion mechanisms where motion reversals, etc., occur under impact conditions, in small fractions of a second. As will be seen later, this results in a great deal of difficulty in estimating maximum stress levels, etc., in such devices. In many cases, however, it is sufficient for the designer to realize that such fillets must exist. He must estimate their possible duration and put them into the measured or derived curves, accordingly.

Another aspect of this, however, would be to remember that in an intermittent motion device everything starts at zero. The output link of the mechanism is at rest, which means, for graphical evaluation purposes, that the initial velocity and acceleration are zero. The initial net force on the body is also at zero to begin with. The previous graph of Fig. 2–14 indicated that the velocity had reached a finite value at time zero and this resulted in infinite accelerations to get started. If a time at which the whole mechanism is at rest is picked as the zero time point, and the fact that it takes a finite period of time for it to start moving is recognized, these difficulties will be avoided.

In graphical mechanics, all calculations begin with everything at rest in order to avoid what the mathe-

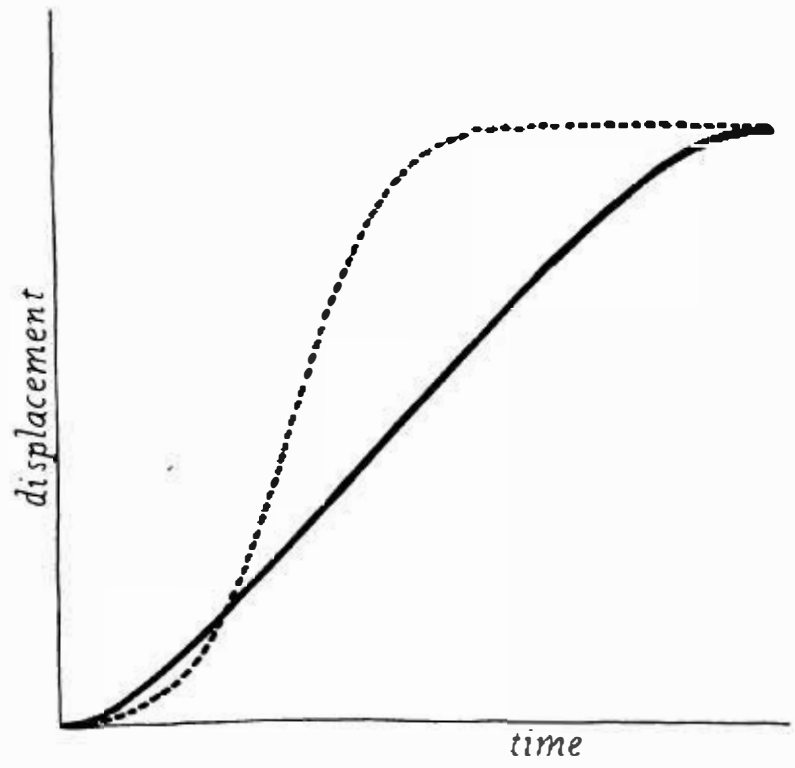


Fig 2–16. Similar displacement curves for an internal Geneva (solid line), and external Geneva (dotted line).

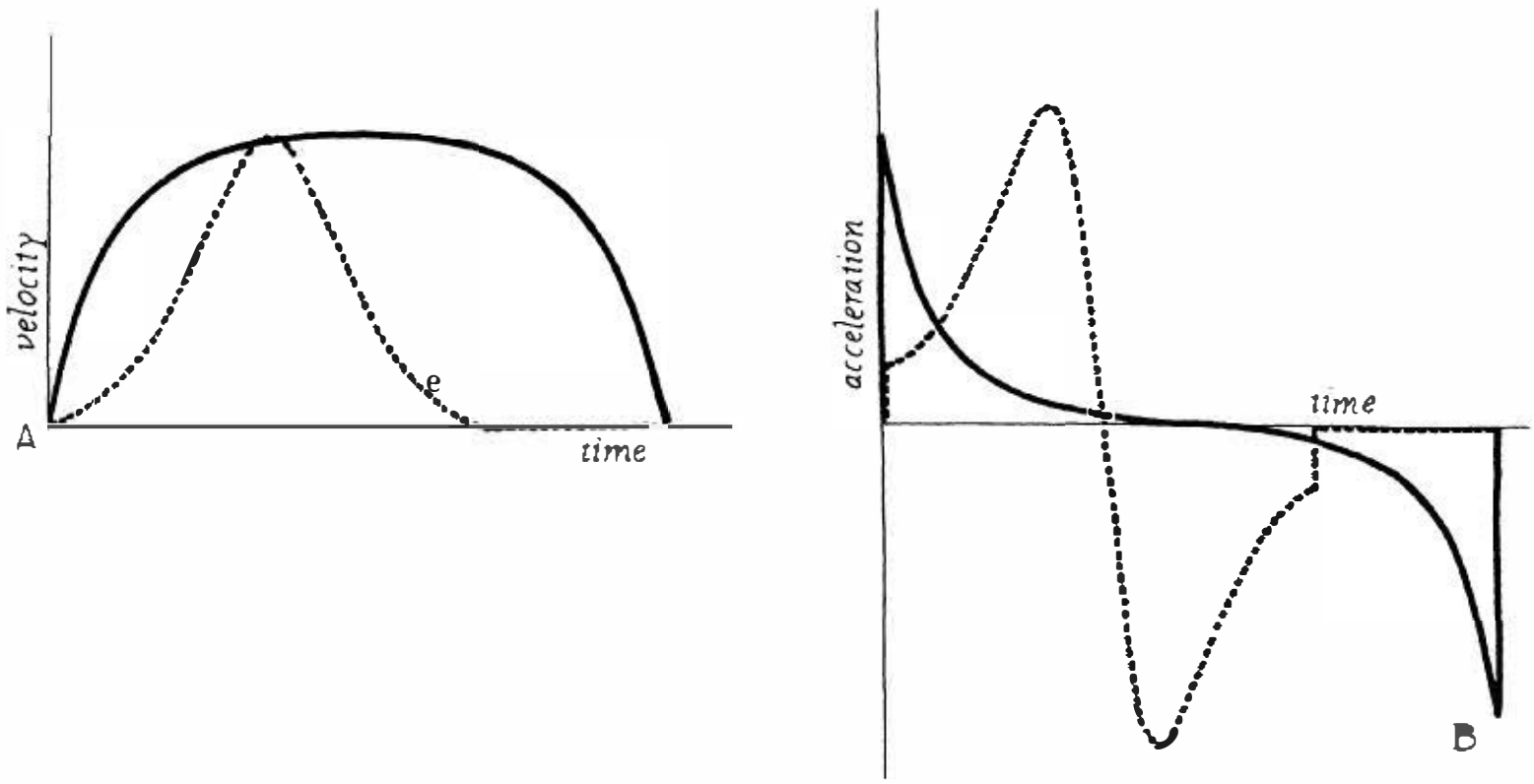


Fig. 2–17. (A) Velocity and (B) acceleration curves for the internal and external Geneva. The curves for each mechanism are drawn to different vertical scales to emphasize their shape differences. (See Fig. 9–6 which shows the same curves drawn to a common scale.)

maticians call "constants of integration," as mentioned earlier. The above results show why this is true.

Another Caution—Watch Out for Small Differences!

In the previous section it was noted that a small difference in the displacement curve (the addition of nearly invisible fillets at the beginning and end of a displacement curve) turned two infinite accelerations into two finite accelerations. Other small changes in curve shape can make a surprising difference in the shapes of related curves. The beginner is advised to be very careful, and to plot more points on his derived curves that he would be inclined to do, at first, until he has had a good deal of experience with graphical methods. Figure 2–16 shows superimposed displacement curves for internal and external Geneva mechanisms (described in detail in Chapter 10). The curves are drawn to different vertical scales (the internal Geneva has lower acceleration than the external Geneva for a given input speed) in order to emphasize their different shapes. Notice that both displacement curves seem to be "ramps" with fillets at the beginning and end of the ramps. The dotted curve shows larger fillets and a steeper slope than the solid curve, but otherwise they seem very similar in basic shape.

Figures 2–17a and 2–17b show the velocity and acceleration curves of internal (dotted curves) and external Genevas. The velocity curves may have a distant relationship to each other, perhaps; although some obvious differences have begun to appear. The acceleration curves though are very different. Small differences between the basic shapes of the original curves failed to suggest the drastic differences in the acceleration patterns of these two similar mechanisms.

Jerk

One more topic must be considered before this discussion of linear motion is completed; that is, the subject of *jerk*. The subject of jerk does not appear in physics books, but machine designers insist that it exists and that it must be considered in design situations involving heavy loads and/or high speeds. Jerk is defined as the rate-of-change of acceleration of a body. As such, it is computed by differentiating (graphically or mathematically) the acceleration-versus-time curve, as shown in Fig. 2–18. Although infinite acceleration would require an infinite force and is, therefore, impossible; infinite jerk simply means "an infinite desire to start in motion once the force is suddenly applied." Although from a purely theoretical point of view, any vertical step in the

acceleration curve produces infinite jerk, designers assure us that a smaller step change in acceleration produces less jerk than a larger step. Machine tool designers worry about this parameter, and we will consider it from time to time as we compare the relative merits of various intermittent motion devices.

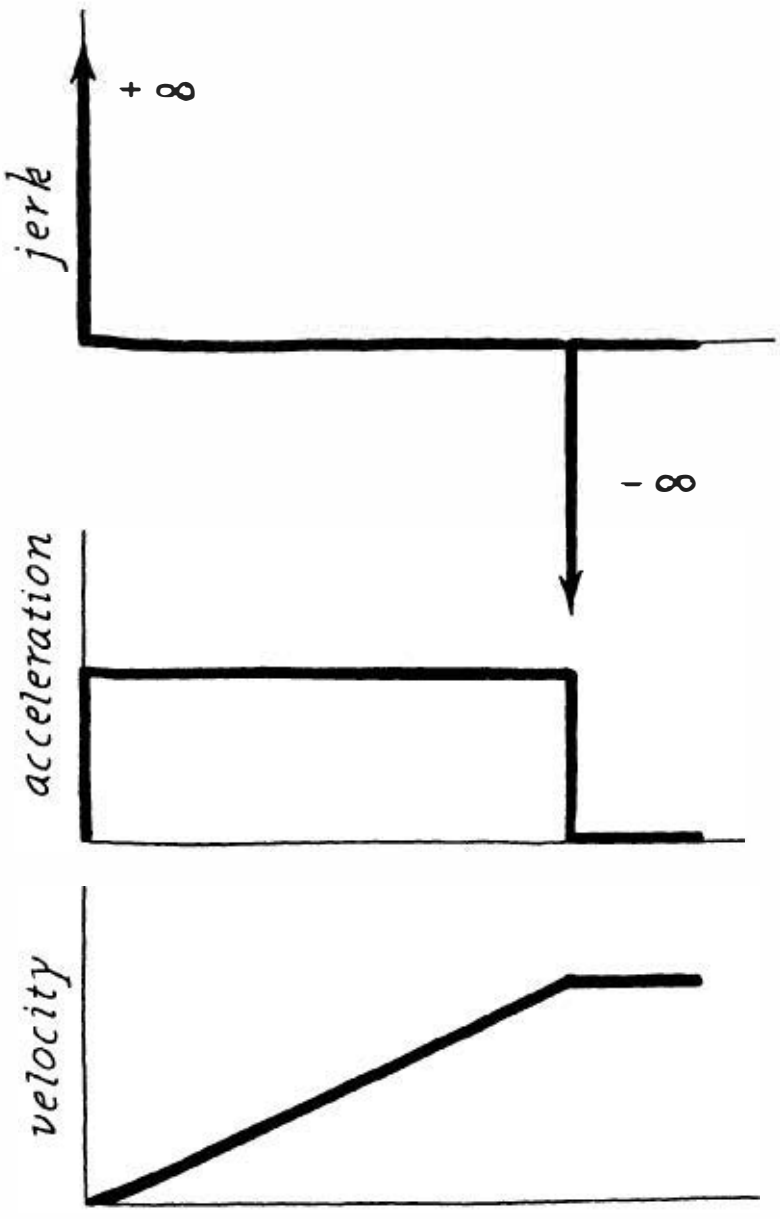


Fig. 2–18. Relationship between velocity, acceleration, and jerk curves.

ROTATIONAL MOTION

Thus far, only the case of linear motion has been considered. In most machine design situations rotational motion will be of far more interest. Fortunately, now that the linear ease is understood, only a new set of terms and units is needed to handle the rotational case.

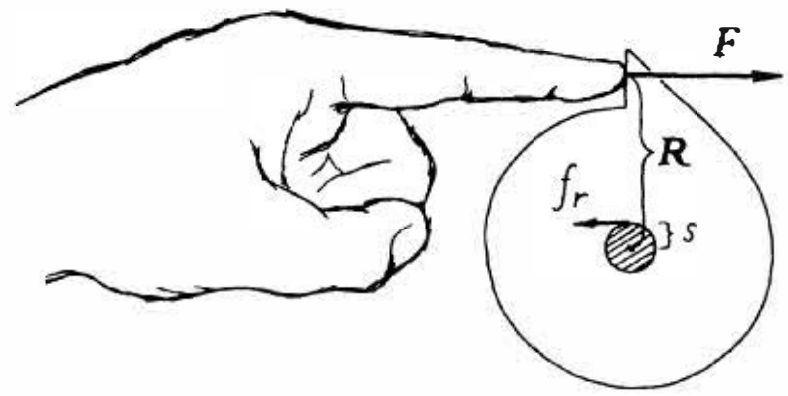


Fig. 2–19. Forces acting on a disc.

In Fig. 2–19, a finger pushes a ratchet wheel. This produces a drive torque on the wheel $F \times R$ and the inevitable friction reaction torque $f_r \times s$. If the finger pushes the wheel briefly and then stops, the wheel will eventually coast to a rest under the action of the frictional torque, as in Fig. 2–20. This is an exact analogue of the block-on-the-table example studied earlier, and the curves which we would draw of torque (τ), angular acceleration (α), angular velocity (ω), and angular displacement (θ), are identical to those drawn for the earlier example. We use different symbols and units for the various factors involved, as shown in Fig. 2–21. Compare this illustration to Fig. 2–12 to see that they are, indeed, exact analogues of each other. We move down in the curve set by taking areas; we move up by measuring slopes; just as before.

The second curve in Fig. 2–21 describes the angular acceleration of the wheel as a function of time and as a result of the applied torque. The angular acceleration curve is found by dividing the torque on the body at any instant, by the moment of inertia of the body; just as, previously, linear acceleration

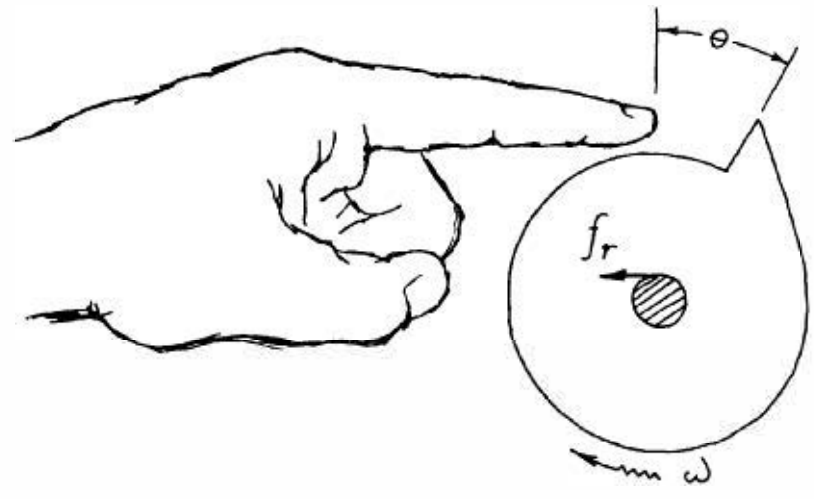
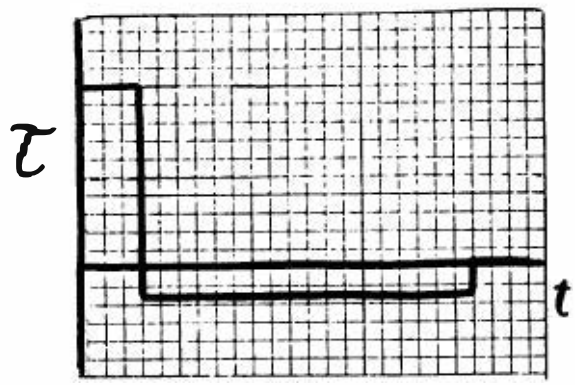
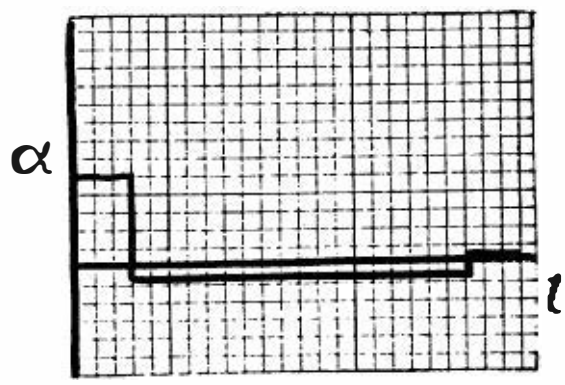


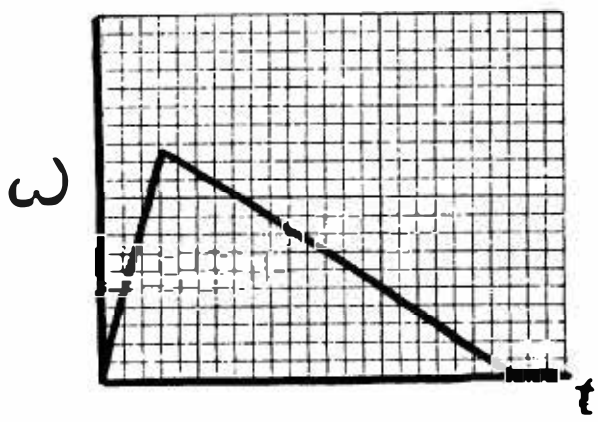
Fig. 2–20. Frictional force acting on a coasting disc.



Divide τ by "moment of inertia," I to get α curve, below, or multiply α by I to get τ curve, above.



Take areas under α to get ω, below, or take slopes of ω to get α, above.



Find areas under ω curve to get θ curve, below, or find slopes of θ curve to get ω curve, above.

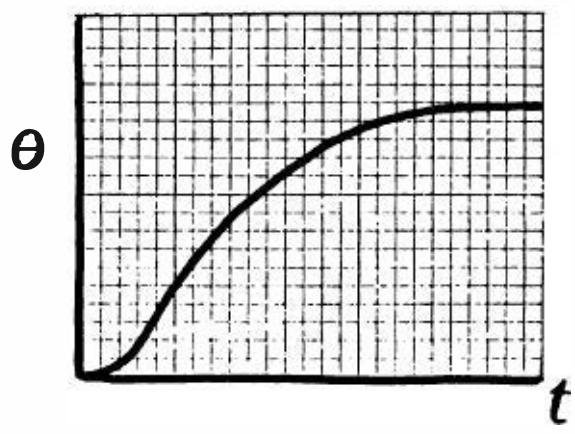


Fig. 2-21. Relationship between torque, angular acceleration, angular velocity, and angular displacement curves for the finger-on-a-disc case.

was found by dividing force by mass. The equation:

$$T = I\alpha$$

is the rotational analogue of $F = ma$.

It should be noted for accuracy's sake that $T = I\alpha$ is only an analogue (a mathematical equivalent, not a physical equivalent) of $F = ma$ and is not, in itself, a basic law. $T = I\alpha$ only applies when a body is constrained by bearings to rotate about a principal axis through its center of gravity. It does not, therefore, describe the complete motion of unbalanced bodies or of free bodies rotating in space. However, to gain an insight into the design problems of intermittent motion mechanisms without attempting to gain exact solutions, this analogue will serve the purpose very well (and most machine bodies obey the above restrictions).

Rotational Units

The unit of angular displacement in the English system is the radian. There are 2π radians in a complete circle, thus one radian is equal to 57.3 degrees. The unit of angular velocity is, therefore, radians per second; since velocity is rate-of-change of displacement. The unit of angular acceleration, similarly, is radians per second per second, or radians per second, squared. And torque comes in lb-ft. The unit of moment of inertia is slug-ft^2 or ft-lb-sec^2. Moment of inertia is a measure of the resistance of the rotating body to angular acceleration. It is discussed at some length in the appendix.

Angular Acceleration

The angular acceleration (α) of a rotating body is its rate-of-change of angular velocity (ω). Exact numerical values for certain points on an angular acceleration curve are often wanted when plotting sets of motion curves. There are two ways to obtain these. If the instantaneous applied torque (drive torque, less friction and load torques, in many cases) on a given body is known, its instantaneous angular acceleration can be found by applying the equation: $\alpha = \tau/I$. Sometimes, friction torque can be ignored, which simplifies things.

Another method is to use the techniques of Chapter 1 to calculate two values of ω at two different times, t_2 and t_1. Since angular acceleration is the rate-of-change of angular velocity it is then possible to use the equation:

$$\text{Average } \alpha = \frac{\omega_2 - \omega_1}{t_2 - t_1}$$

to compute the average value of α in the time interval, t_1 to t_2. If t_1 and t_2 are close to each other, there will be little difference between average acceleration and instantaneous acceleration for that interval of time.

To be complete, it should be mentioned that there are other accelerations associated with rotational motion; such things as Coriolis acceleration and radial acceleration. These account for the changes in tangential (linear) velocities that occur as an object rotates, or moves to a different radius. However, at this point, our concern will be only with angular acceleration: the increase or decrease in the number of radians per second of angular velocity of the body.

Angular Displacement

Frequently, a set of motion curves is started by plotting angular displacement versus time; differentiating this graphically, to derive velocity and then again to obtain acceleration curves. The displacement curve is produced with the aid of mylar or paper templates and the original design drawings; determining output displacement as a function of input displacement, perhaps, and then converting this to a time-based curve by using other design information such as the speed of the drive motor.

OTHER ASPECTS OF GRAPHICAL MECHANICS

How to Handle More than One Force

The force plotted in studying linear motion was the net horizontal force on the body. In most machine design problems interest will be in the motion of a body under the action of a single resultant force or of some component of the resultant force, as discussed in Chapter 1. Fortunately, the methods already derived can be used to determine the response to either a resultant or to a component. How can the complete motion of a body be analyzed by evaluating its motion for several components separately?

The diagram at the top of Fig. 2–22 shows a hand that is pushing a block along a book on a table. The hand exerts a positive driving force of F_d on the block. The book exerts a negative friction force of f_f, as shown in the illustration. At time, $t = 0.1$ sec, the block runs off the edge of the book, but the hand keeps on pushing it in the same direction until it lands on the table where it eventually slides to a halt. This might be a little difficult to do, but it definitely would be possible.

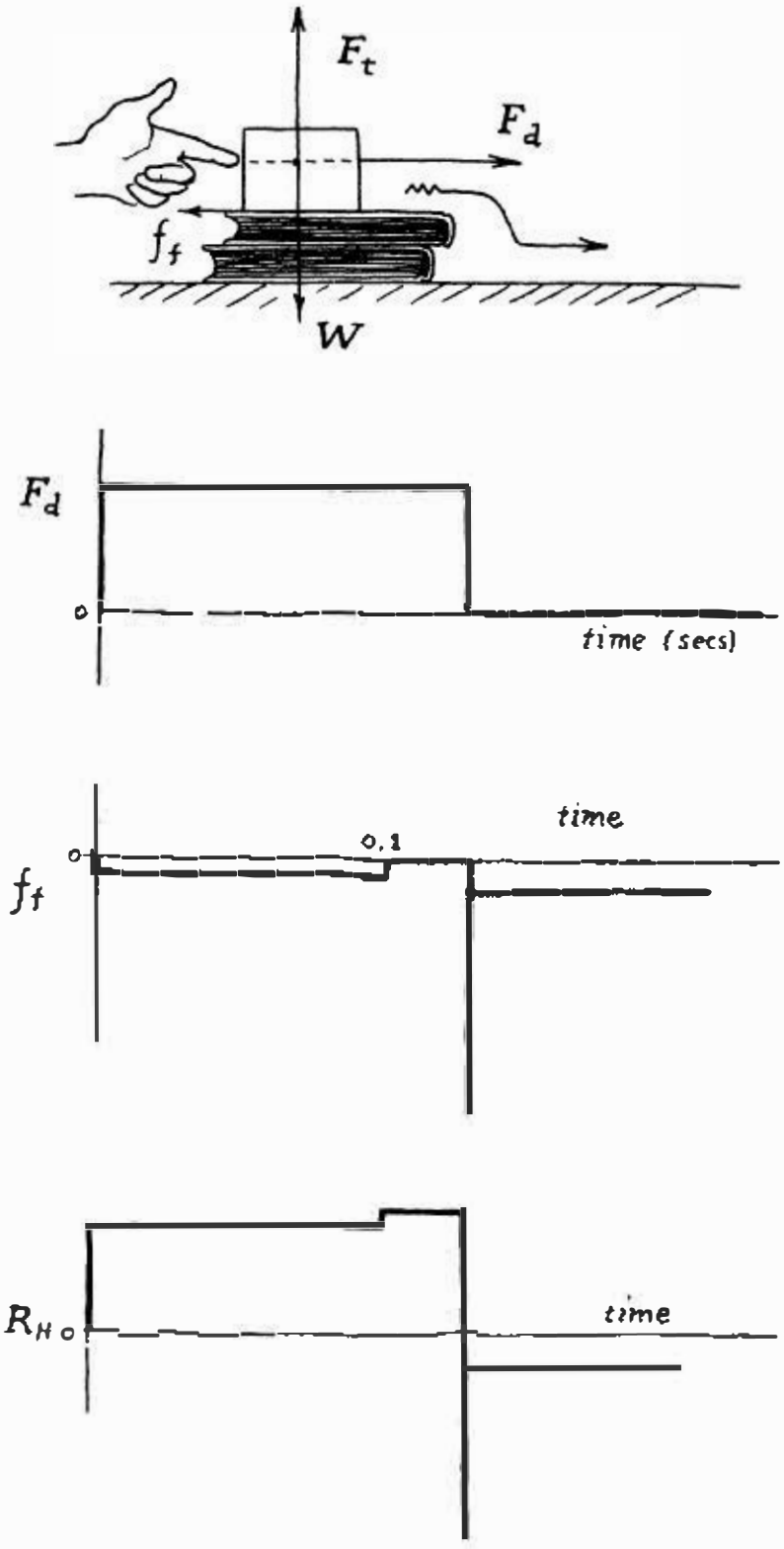


Fig. 2–22. Forces acting on a block which is being pushed off a pile of books; and graphs of the component and net horizontal forces on the block.

In the graphs below the diagram, in Fig. 2–22, two different horizontal component forces are plotted; the force of friction and the force exerted by the

finger on the block, also, the net horizontal resultant (which was used in the original block-on-a-table analysis to determine the horizontal acceleration, velocity, and displacement of the block as a function of time). The negative spike in friction force occurs when the block first strikes the table and impact increases the normal force on the table.

Plotting finger and friction forces separately in this manner, can be useful. This allows for estimating the amount of work done against friction; for example, the impulse imparted to the block by the finger, etc., as will be seen later on. Finger and friction forces, furthermore, would be easier to measure on an experimental basis than would be the resultant. They could be measured and plotted separately, and then added graphically, to derive the resultant, if this were the case.

There are also other forces on the block which are not considered at all by the analysis just completed. While the block is sitting or sliding on the book, the book is exerting a vertical supporting force against it, labeled F_t, in Fig. 2–22. At the same time, the weight of the block is pressing downward against the book with an equal and opposite force (W). Once the block has been pushed over the edge of the book, the supporting force F_t, vanishes, and the only remaining vertical force is W, the weight of the block (ignoring any vertical frictional component between the finger and the block).

The graph in Fig. 2–23 shows F_t, F_b, W, and the net resultant vertical force R_v as a function of time. This resultant vertical force is 0 while the block is on the table, but becomes finite (and negative) once the block has gone over the edge of the book. The block at this point, then, will start to move (accelerate) in a downward direction under the influence of this unbalanced vertical force. Finally, it strikes the table which stops it with a vertical blow and then supports it with a force F_b, again equal and opposite to W. Figure 2–23 shows the entire process.

By using the previous procedures, both horizontal and vertical accelerations, velocities, and displacements of this block, could now be determined, considering the horizontal and vertical motions separately.

Once horizontal and vertical values for acceleration, velocity, and displacement are determined, nothing further is needed to predict the total position, motion, etc., of the body in the XY plane as a function of time.

In fact, total motion can be found in this case

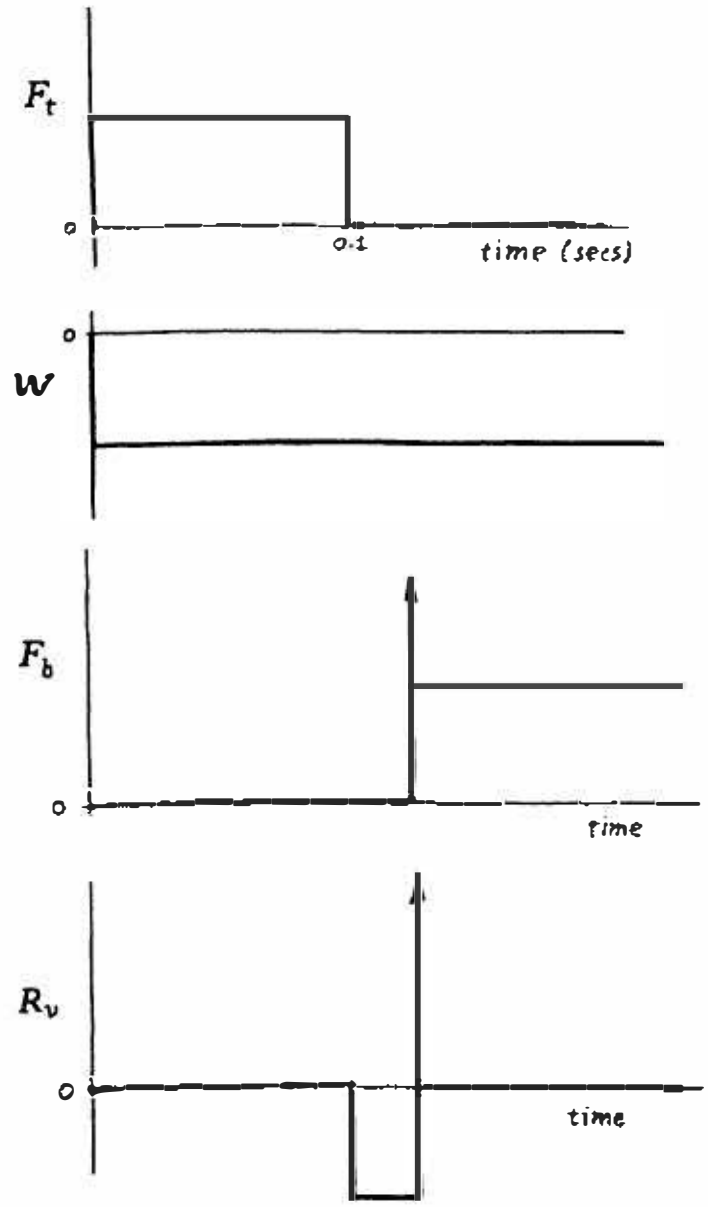


Fig. 2–23. Graphs of component and net vertical forces on the block being pushed off a pile of books.

only by calculating horizontal and vertical motions separately, then using these results as X and Y components of acceleration, velocity, and displacement to plot the total motion of the body. It is possible, with vector mathematics, to handle single body resultants that change magnitude and direction, but this cannot be done with the graphical techniques described here. The graphs *only* show variations in *magnitude* of vector quantities whose line-of-action directions do not change. (The vectors can reverse direction by 180 degrees, go from positive to negative, but the line of action must not change.) The resultant force on the block on a book, the single force that sums up all the X and Y forces on the block, starts as a pure X force, when the weight of the block is supported by the books; turns into an XY force as the block leaves the books; and ends

as a pure X force as the block slides along the table. Each of these resultants could be plotted separately to find the total motion of the block, but "initial velocities," etc., would have to be accounted for at each switch from one resultant to another. This could cause serious trouble, as explained earlier. Thus it is necessary, when resultant forces change directions, to stick to components when making motion curves. In the first two examples, i.e., the original block on the table and the rotating disc, the motion can be analyzed with either components (separating friction and finger forces, for example) or resultants (adding finger and friction forces and figuring the response to this resultant), because the line-of-action direction of the resultant force in one case, or torque in the other, does not change.

Work, Energy, Momentum, Etc.

The science of mechanics deals with other things besides force, acceleration, velocity, and displacement, of course. Work, energy, impulse, and momentum, for example, are commonly used concepts and are all useful for mathematical solutions of many problems. It will not be necessary to examine all of these concepts in depth, in this book, since the goal of achieving a rough understanding of "what goes on" in intermittent motion mechanisms can be attained even though practice is restricted, primarily, to the FAVD approach.

Furthermore, concepts such as energy and momentum are more difficult to handle, graphically, than FAVD because they involve simultaneous treatment of several bodies in a machine. In FAVD a free body is isolated and studied alone, although, of course, it is necessary to know what forces other elements of the machine exert on the body being studied. In dealing with energy, on the other hand, laws such as that for the conservation of energy, which says that the total energy of a multibody system remains constant unless acted upon by external forces, must be respected. This means that during collisions or interactions between bodies, energy within the system is redistributed or altered in form; with potential energy in one body, for example, being converted to kinetic energy in another; with kinetic or potential energy being converted to useful work; or with mechanical energy being converted to heat energy and being lost as far as useful output is concerned. These are powerful concepts, but again, are really beyond the scope of this work and are unnecessary for our purposes. Included in Chapter 3, however, is a brief discussion of impulse, momentum, energy, etc., as related to impacts between machine members.

Table of Curves—and Some Basic Rules for Constructing Curves

Students of calculus make frequent reference to tables of differentials and tables of integrals when trying to do mathematically, what has been done here graphically. It would seem logical, therefore, to include in this text a pictorial table showing the relationships between different types of curves that will be encountered in many intermittent motion devices. Once the student learns to recognize these different shapes he can often integrate and differentiate curves almost by inspection.

First, however, some basic relationships between curves should be considered. Figure 2-24 shows the relationship between a zero slope on one curve, the axis crossover point on the related "derivative curve," and the change in slope on the related "integral curve." This relationship helps us find maximum

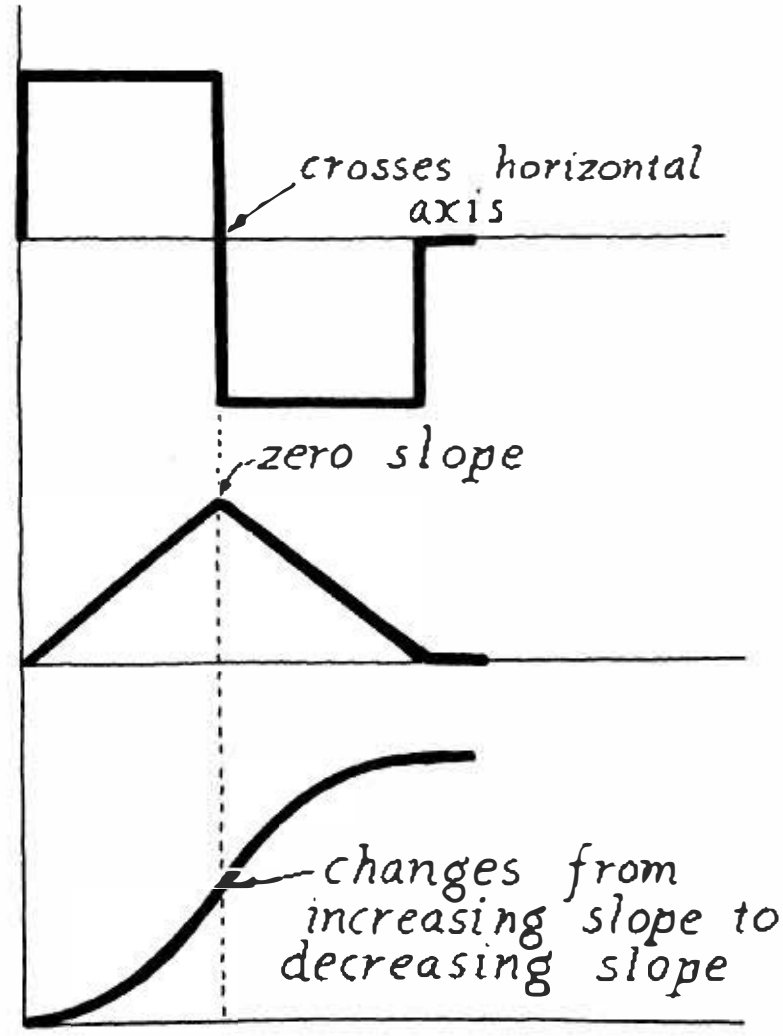


Fig. 2-24. The relationship between, the axis crossing, and the zero slope in a series of related curves.

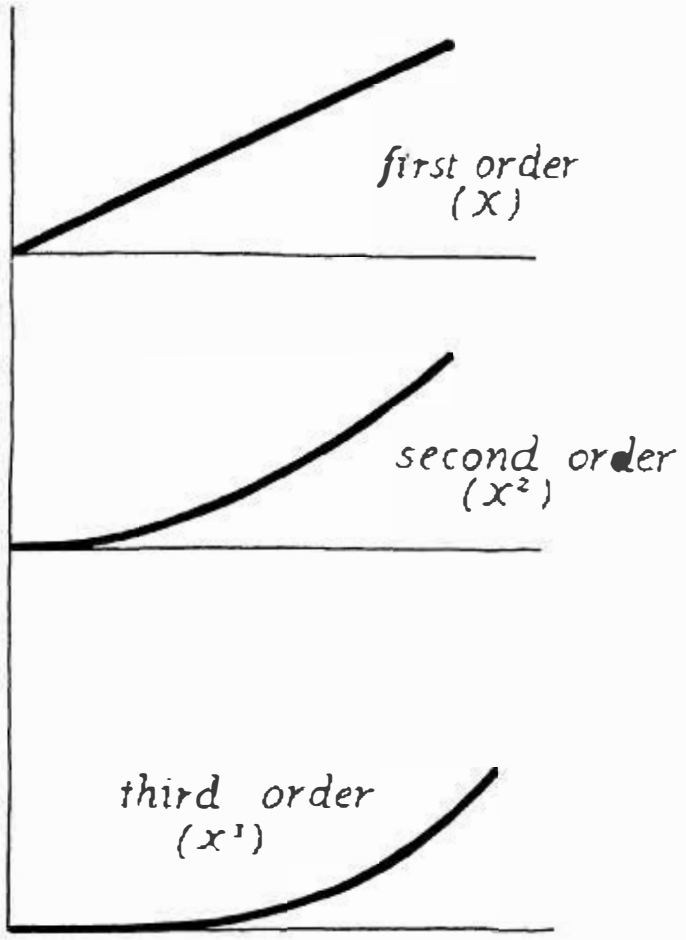


Fig. 2-25. Curves become more concave with successive integrations.

and minimum points on "integral curves." Figure 2–25 shows how the "order" of the curve (its degree of concavity) increases with multiple integrations (moving down): or decreases with multiple differentiations (moving up). Figures 2–26 and 2–27 show

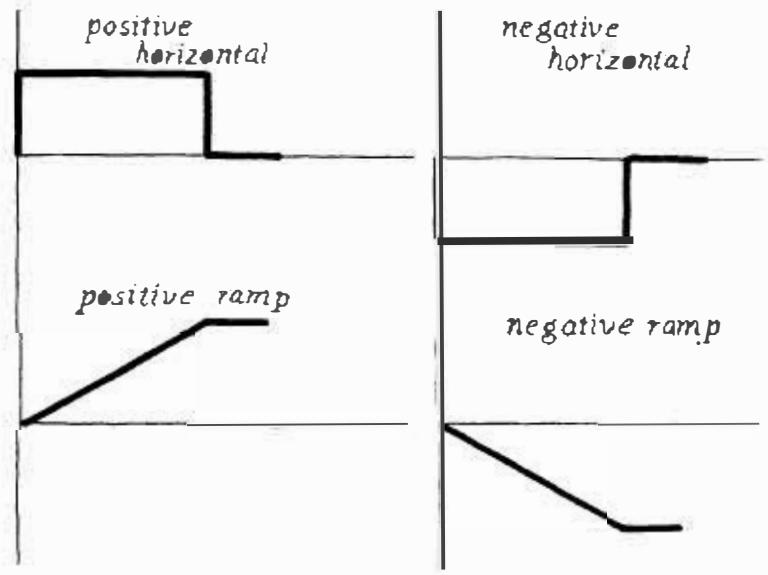


Fig. 2-26. Positive and negative curves of the same type.

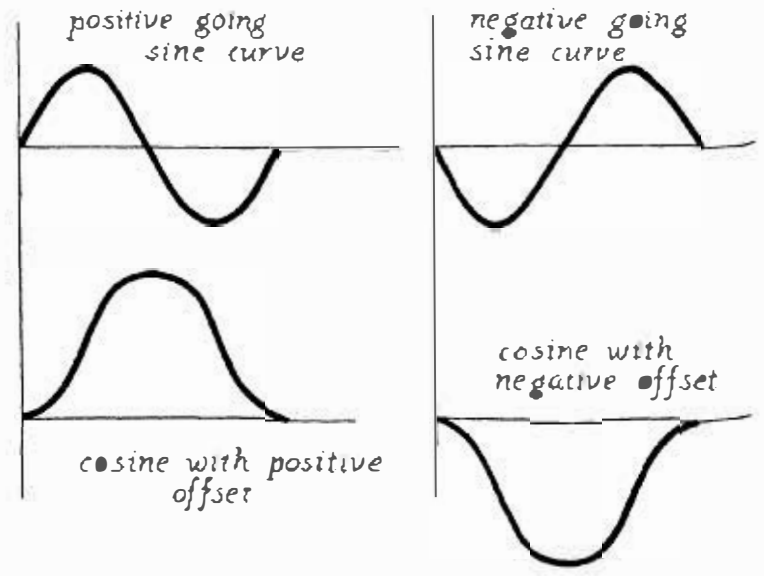


Fig. 2–27. More positive and negative curves.

how mirror-image curves produce mirror-image integral, or differential, curves.

When one basic shape of curve follows another, sequentially, in a single curve, the integrals and differentials also follow, with each portion of the derived curve being that which would be obtained for the related portion of the original curve if taken alone. For example, in Fig. 2–28a, the integral (going

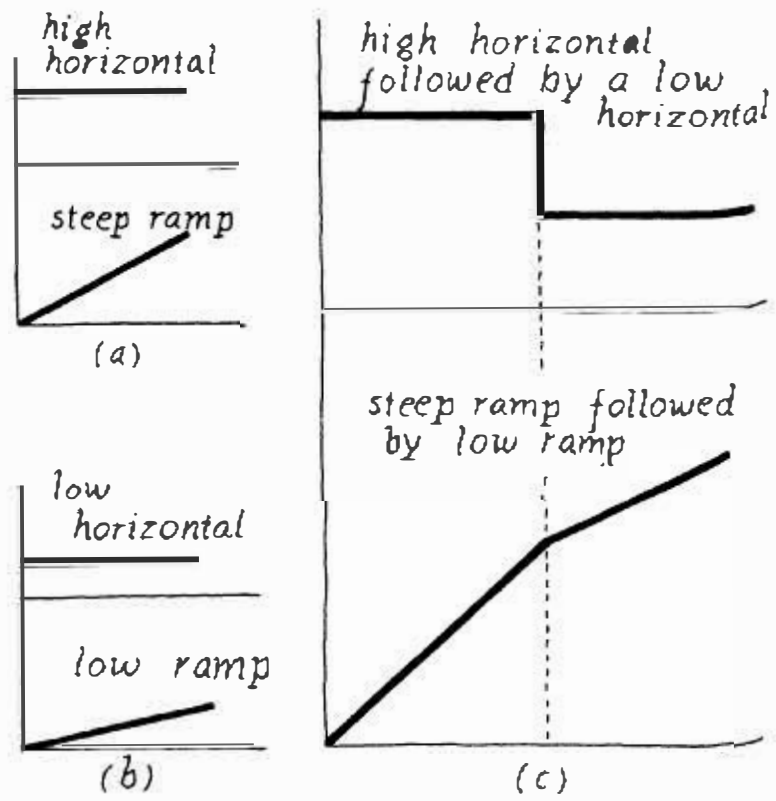


Fig. 2-28. When one type of curve follows another, the integral or differential curves also follow in the same order.

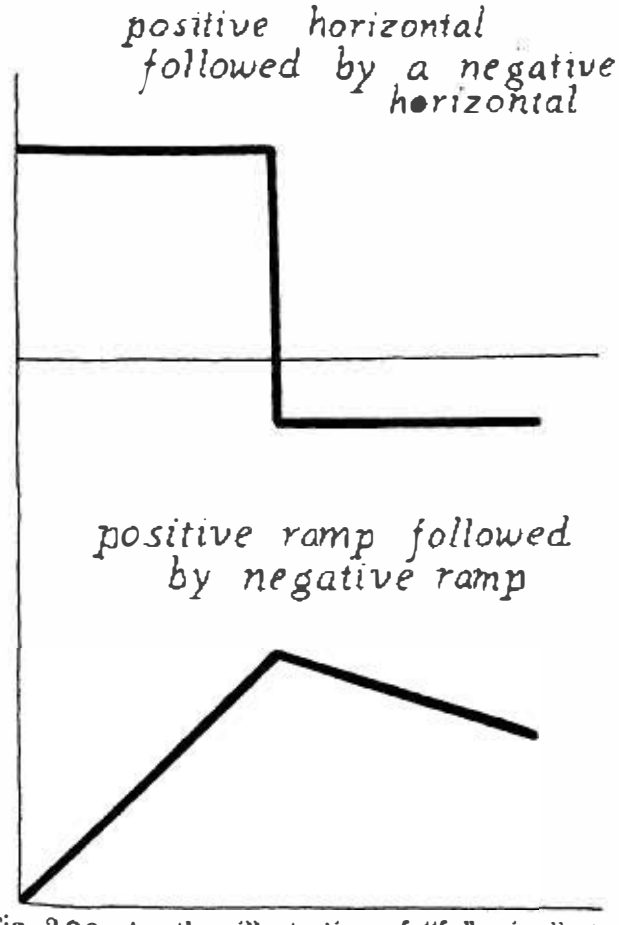


Fig. 2-29. Another illustration of "following" curves.

down) of a high horizontal curve would be a steep ramp. The integral of a low horizontal, Fig. 2–28b, would be a low ramp. If there is a single curve, as in Fig. 2–28c, in which a high horizontal is followed by a low horizontal, its differential curve will be a steep ramp followed by a low ramp. Notice, however, that the low ramp starts at the end of the high ramp rather than on the horizontal axis.

Similarly, in Fig. 2–29, a positive horizontal, followed by a negative horizontal, produces (going down) a positive ramp followed by a negative ramp.

Sometimes, curves are superimposed or added together, rather than following each other sequentially. In Fig. 2–30, for example, are a cosine curve and its integral, Fig. 2–30a, a horizontal and its integral, Fig. 2–30b; and an offset cosine curve (a cosine curve superimposed on a horizontal, as shown by the dotted line) and its integral (a sine curve superimposed on a ramp).

These are some of the "rules." There are undoubtedly others. Nothing beats experience, however. By sitting down and counting squares or measuring slopes to derive one curve from another, a feel for graphical mechanics is soon achieved. Nothing else really needs to be understood except:

(a) How to get that first curve of F, or A, or V, or D versus time
(b) How to measure areas
(c) How to measure slopes.

The rest follows automatically.

A series of ten drawings makes up Fig. 2–31, to show the relationships between common curve sets which are often found in machine design.

Figure 2-32 shows Da Vinci's famous perpetual motion machine. With a knowledge of how to "calculate" intermittent motion mechanisms, it should be possible to discover why it did not work.

In this device, a large ratchet wheel has a hub on which are mounted twelve arms. Each arm is free to rotate through an angle of approximately 60 degrees about a pivot in the surface of the drum. There is a weight at the outer end of each arm, and a series of stops on the outer ratchet wheel which catches the weights on the rim in the positions shown.

Presumably, it is necessary to give the wheel a shove to start it in motion. Once in motion (in a counterclockwise direction) it is supposed to keep

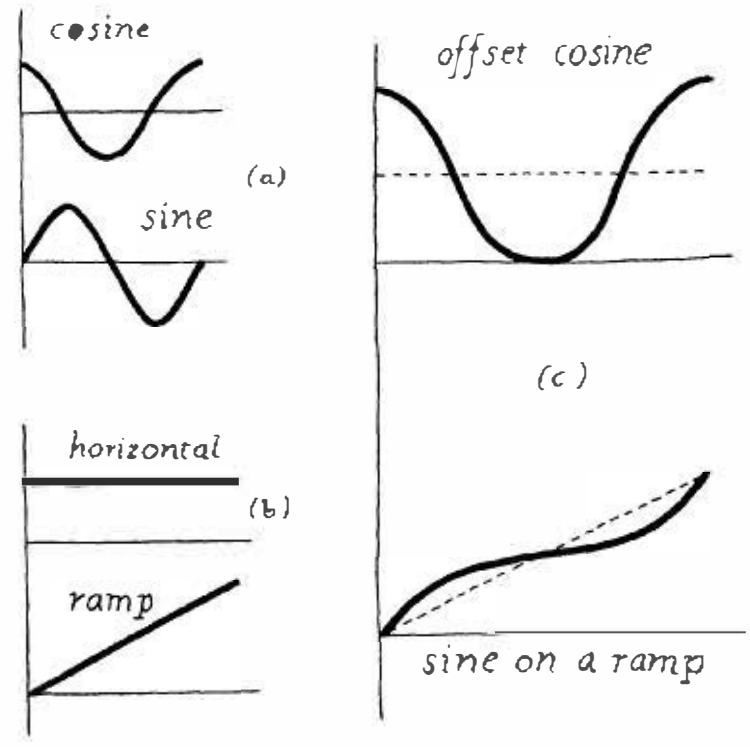


Fig. 2-30. Curves (and their integrals) can also be added. An "offset cosine" is really a cosine added to a horizontal. The integral curve, sine on a ramp, is the sum of the integrals of cosine and horizontal.

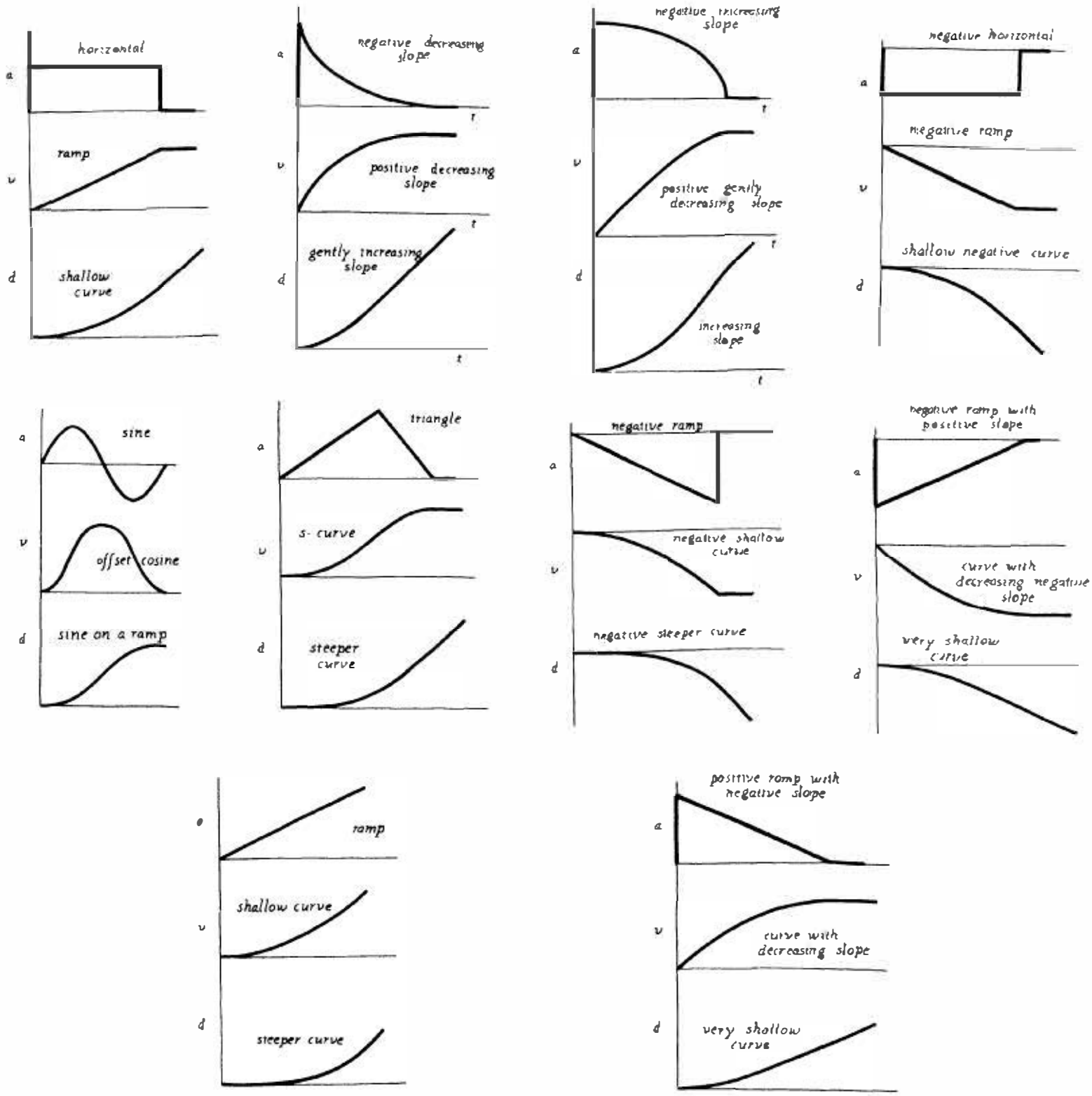


Fig. 2-31. Graphical Table of Integrals and Differentials. Shows relationships between common curve sets frequently encountered in machine design.

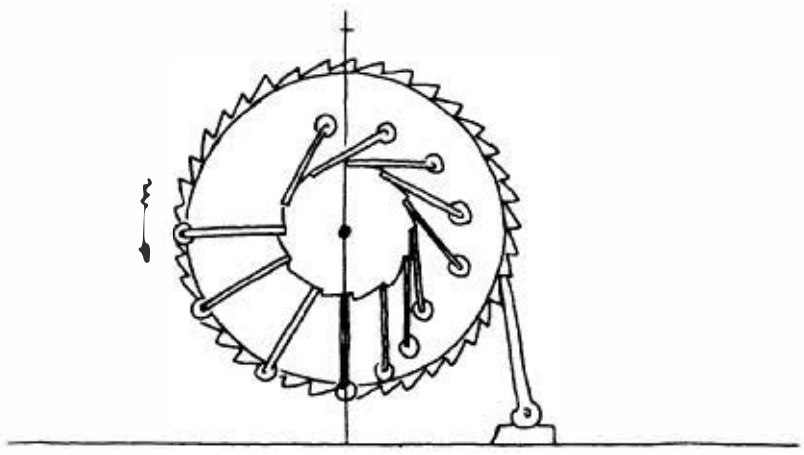


Fig. 2-32. Leonardo Da Vinci's perpetual motion machine, one of the earliest intermittent motion type designs.

itself going. As the wheel turns, the arms are lifted, as shown, over the top of the drum until they are out of balance and fall (rotate) until the weight on the end of the arm strikes the rim of the ratchet wheel. Leonardo presumably hoped that the fact that the weights are falling through a greater radius than they are being lifted would give his wheel a net torque and produce perpetual motion. A "no-back" pawl has also been provided, however, which indicates he knew better, or at least, he wasn't sure! It is not really an intermittent motion device, but is certainly an early and ingenious use of a ratchet wheel, which we will be studying at length, later on.

CHAPTER 3

Elastic Body Mechanics

ELASTIC BODIES

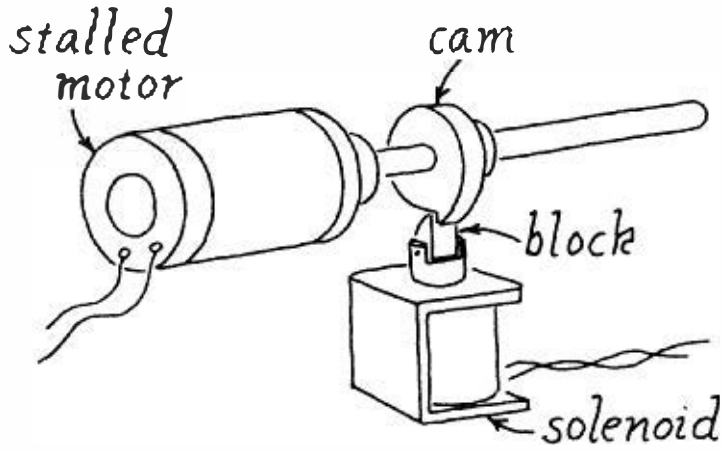
Rigid Body Mechanics Not Sufficient

Thus far, only conventional, or rigid body mechanics has been considered; the type learned daily by undergraduates. But it is also the kind that fails to solve real-life problems which occur in dealing with many types of intermittent motion mechanisms, or any high-speed mechanism that involves a lot of clearance, backlash, and/or sudden changes in velocity. To see why rigid body mechanics is not always sufficient, consider the action of a simple escapement mechanism Figure 3-1 shows a shaft that is continuously urged to rotate by a stalled electrical motor. A cam on the shaft contacts a block which normally prevents rotation of the shaft. When the block is momentarily pulled out of the way by a solenoid the shaft will turn. After one turn the cam strikes the block and output motion ceases. (The solenoid, of course, has to be de-energized soon enough to let the block return to the interference position before the cam completes its first revolution, if only one turn is desired.) Not a very elegant mechanism, but it does produce intermittent motion, and it is very simple.

The torque-time curve for the control cam of this escapement is shown in Fig. 3-2. When the block is withdrawn, the motor produces a torque on the cam and shaft, and these members then will move (with steadily increasing velocity). When the cam strikes the re-positioned block, however, it must stop instantly to conform to the fact that the block has zero velocity. (Or the block must suddenly acquire the velocity of the cam, which the designer presumably does not want.)

But as was seen in the previous chapter, an instantaneous stop would mean an infinite negative acceleration, hence, an infinite torque. These things, however, are just not possible What actually does happen, is that each of the colliding bodies, the cam and the block, try to alter their respective velocities to conform to that of the other. They ultimately succeed. The length of time this "getting together" takes, and the forces that are generated during this process depend on such factors as the stiffness, inertia, and impact velocities of both bodies and, to a lesser extent, upon the frictional and drive torques which are acting on the bodies during the collision. It seems a little strange to have to say "to a lesser extent" about something as tangible as a drive force, but most practical drive forces are so small in comparison to actual impact forces that they have little influence on impact duration times or peak forces.

The impact forces that are generated in collisions of this type get to be extremely high, even in small, lightweight instruments driven by what appear to be relatively weak drivers. In most machine applications, in fact, the forces that are generated as the bodies struggle to conform to each other's desires are so high as to deform the bodies. Once this happens, however, we are dealing with elastic body mechanics instead of rigid body mechanics.



Drawing Courtesy of MACHINE DESIGN Magazine; Nov. 21, 1968

Fig. 3–1. Simple electromechanical intermittent motion mechanism.

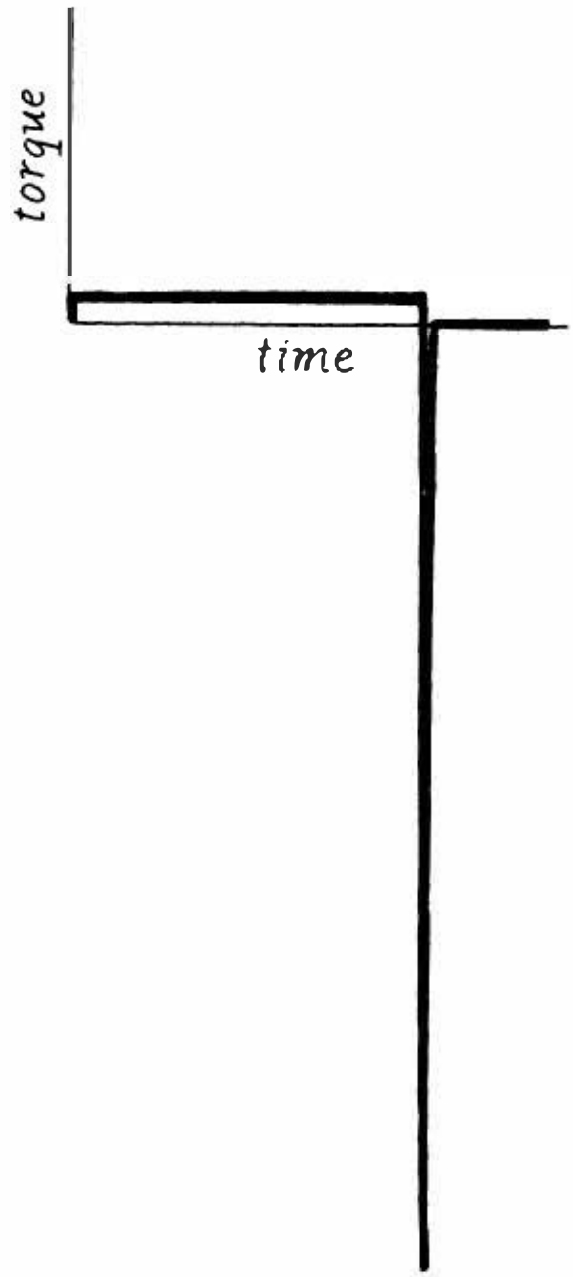
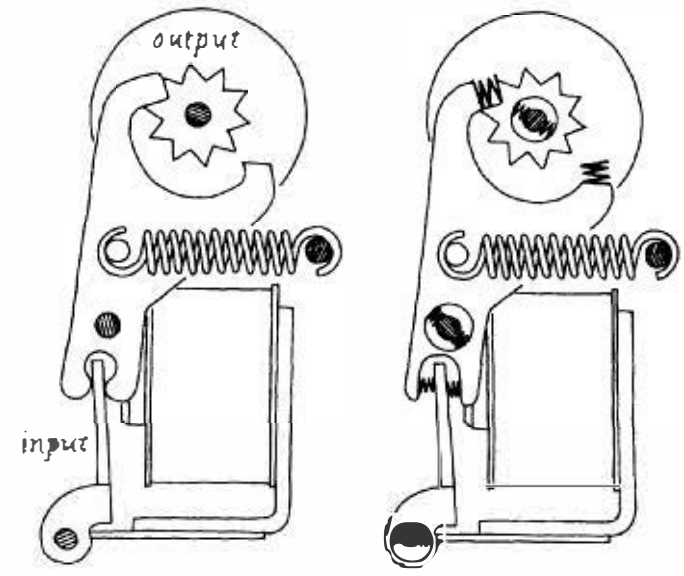


Fig. 3–2. Torque versus-time curve for the cam of Fig. 3–1.

The motions of machine bodies and the forces which they exert on each other, as well as the internal stresses within the bodies, are all modified by the fact that the bodies are now deforming as they operate. Each link in a mechanism train behaves like a spring; storing, releasing, and dissipating energy. Designers have found from experience that if they try to treat the bodies as pure springs for computational purposes, they will get into all sorts of complications and confusions. In most design situations it is sufficient, instead, to find a substitute "model" for the body which is easier to understand than a pure spring and which is easier to handle from a theoretical standpoint. One of the most useful of such models is called the *quasi-static model.*



Drawing (Left) courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965: p. 181F

Fig. 3–3. (*Left*) Inverse escapement. (*Right*) A quasi-static model of the same mechanism.

The Quasi-Static Model

Machine members can be thought of as rigid bodies connected by small springs, as shown in Fig. 3–3. The masses in this model are assumed to be perfectly rigid and to obey all of the laws of rigid body mechanics. All elasticity of the machine members is assumed to lie in small springs that connect the rigid masses together. Mathematicians call this a quasi-static model of elastic bodies and find it very useful in approximating the performance of elastic bodies in actual design situations.

In the official quasi-static model the springs are assumed to be linear; that is, if the forces acting on the springs are doubled, their deflection is dou-

bled. They are also assumed to be "perfect" springs, which means that no energy is lost within the spring as it deforms. This latter is necessary in order to simplify the equations used to study such bodies.

However, where an approximate understanding only, is sufficient, the fact that some energy is lost during every collision or interaction between two members of a machine can be assumed. As in the model in Fig. 3–3, therefore, each of the various machine links is taken to be rubbing on a frictional surface as it moves. These fictitious surfaces do not confine the motion of the linkage; they merely remind us that energy is being lost within or between these various elements of the machine as they perform their jobs.

For completeness, it should be added that engineers have built other elastic body "models" found to be more accurate for certain situations than the quasi-static model. One of these, for example, takes account of the fact that permanent deformation of one or both bodies ean occur during certain types of impact. It is desirable of course, to avoid permanent deformation of machine members and thus, I think that particular model can safely be ignored.

Another model assumes that the springs connecting the bodies are not normal springs, but have nonlinear characteristics; the force required to deflect the spring is not just a linear function of the deflection. This situation is clearly beyond the scope of this text, and besides, the point is to get approximate understanding and not exact answers, in any case.

Using the Quasi-Static Model

First, let us see how to usc the quasistatic model to estimate the torque exerted on the cam when it strikes the block in Fig. 3–1. Assume that the motor exerts a net drive torque of 0.5 ft-lbs on the cam, and that it takes the cam one second to complete its first revolution. Assume also, that a limit switch, set next to the block, turns off the motor the instant the cam strikes the block to avoid complicating the calculation with drive forces. As will be seen later, this is not a bad assumption since the impact forces will he much higher than the drive forces.

Assume next that the cam is brought to rest by the block in 10 milliseconds. This is not a bad assumption in a typical machine design impact; but later on, shorter and longer times will also be considered. Finally, assume that the inertia of the cam, cam shaft, motor rotor, and everything else being brought to a stop by the block is 0.5 slug-ft^2.

The positive area under the torque–time curve of Fig. 3–2 is:

$$\tau_{drive} \times t_{drive} = 0.5 \text{ ft-lbs} \times 1 \text{ sec}$$

or 0.5 ft-lbsec. In order for the block to stop the cam it must exert sufficient torque for sufficient time on the cam to produce a negative torque-time area equal to the positive torque–time area as was seen when studying the finger-pushing-a-block case in Chapter 1.

A stopping time of 10 milliseconds has been assumed, then as a first approximation, merely divide the positive area of the torque-time curve by the stopping time to get the average stopping torque.

$$\text{Positive area} = \text{Negative area}$$

or:

$$\text{Drive torque} \times \text{Drive time} = \text{Stopping torque} \times \text{Stopping time}$$

or:

$$\text{Stopping torque} = \frac{\tau_{drive} \times t_{drive}}{t_{stop}} = \frac{0.5 \text{ ft-lbs-sec}}{0.010 \text{ sec}}$$

$$\text{Stopping torque} = 50 \text{ ft-lbs (average)}$$

This is not a very good approximation, however, because it assumes that the stopping force is constant during the entire stop. The quasi-static model suggests that during the collision the surfaces of the bodies act like springs. The fact is that it takes a very long, soft spring to produce a "flat rate" that is independent of deflection. As most springs deflect, the force required for further deflection increases. And, of course, most machine members are very stiff; they have very high spring rates and resist deflection vigorously. A tiny deformation (fractions of a thousandth of an inch) requires extremely high forces (thousands of pounds) in actual designs involving "rigid" bodies.

If it is assumed that the actual deformation of the cam and block of Fig. 3–1 involves stiff, linear-spring-rate members as suggested by the quasi-static model, then the build-up of stopping force as a function of time will be sinusoidal as in Fig. 3–4, rather than a "square wave" as in the initial assumption when constant torque was assumed during the entire stop.

Note that the torque-time curve is sinusoidal, not triangular. Most designers think of springs as producing triangular curves—and, indeed, they do produce triangular torque (or force) versus *displacement*

curves. But the torque-time curve for a linear spring is always sinusoidal.

The block must still produce the same average 50 ft-lbs of torque if the cam is to be brought to rest in ten milliseconds, however. If the relationship between the average and peak values for one-quarter cycle of the sine wave is known, a factor of proportion can be found that will allow an estimate of the actual peak force that is a little more accurate than was found at first. This factor of proportionality turns out to be about 0.63. (This can be determined graphically by drawing one quarter of a sine wave and computing its average height above the horizontal axis.) The peak torque during the impact now appears to be:

$$\text{Max. torque} = \frac{\text{Average torque}}{0.63} = \frac{50 \text{ ft-lbs}}{0.63}$$

$$\text{Max. torque} = 79 \text{ ft-lbs}$$

or approximately 80 ft-lbs. And all of this has been generated by a ½ ft-lb driver acting for only one second on a ½ slug-ft² cam. Now it can be seen why it was quite safe to ignore drive torque when calculating stopping torque.

The cam has now been brought to rest in the process. However, the small spring assumed to exist between the colliding surfaces has been "loaded." Since it is assumed that the bodies are elastic, the motion stops only for an instant. The spring will now push back on the cam as it unloads, completing the sinusoidal curve started during the first half of the collision (Fig. 3–5). In other words, the cam bounces off the block. If the bodies were fully elastic, the cam would rebound with a velocity equal (but opposite) to the velocity it had upon colliding with the block. In actual machine design situations, however, the rebound velocity will be only a small percentage (perhaps 10 or 20 percent) of the initial velocity; the rest of the impact energy will be lost in internal and external friction during the collision (and the final portion of the curve in Fig. 3–5 would be altered to reflect this).

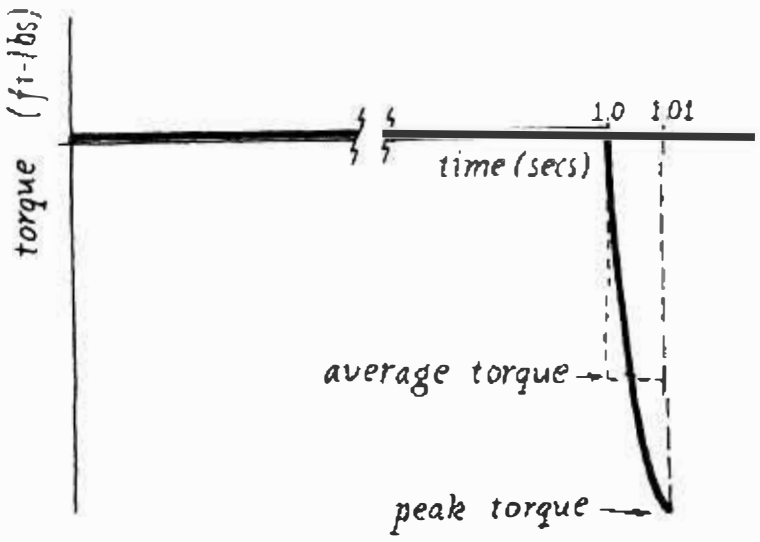


Fig. 3–4. Detail: average torque versus peak torque for the mechanism shown in Fig. 3–1.

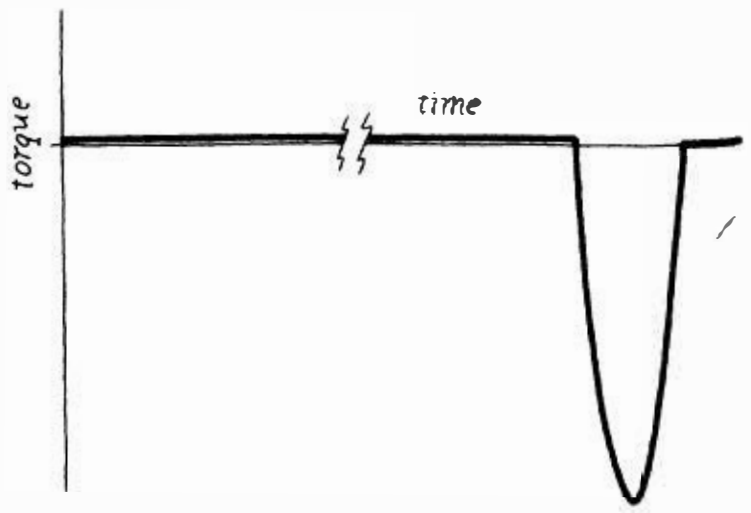


Fig. 3–5. Detail: complete sinusoidal torque-versus-time curve for the mechanism of Fig. 3–1.

We seem to be getting closer to an "exact solution" for the maximum stopping force in this case; but it must be remembered that we started with the arbitrary assumption that the cam was brought to rest by the block ten milliseconds after they touched. If the actual time was 100 milliseconds (which is probably longer than the impact times involved in most machine body collisions) the peak torque would be only 8 ft-lbs. If, on the other hand, the cam were brought to rest in one millisecond (which is in the ballpark for some machine collisions), the peak torque in this same situation would be 800 ft-lbs. Thus, the initial assumption of an impact-duration time has a large bearing on the "answer" obtained. But, hopefully, the graphical process has provided some insight, at least, into what is important and what is not.

Another Point of View

Sometimes we do not know what forces or torques produced the motion of impacting bodies, nor for how long such forces were acting. All we have are experimental data on the relative velocities of the bodies (in the previous example this would be the velocity of the cam) prior to impact. Since we know that impacting bodies can be treated as quasi-static

models (rigid bodies connected by linear springs), we also know that the velocity of the body will decrease from its initial or start-of-collision value to zero, sinusoidally. It is possible, therefore, to draw an angular velocity-versus-time curve while once again assuming a stopping time of 10 milliseconds. And once a velocity-versus-time curve is established, the techniques of Chapter 2 can be used to derive an acceleration curve by graphical differentiation; and then a torque curve can be computed by multiplying the acceleration curve by the inertia of the cam (and the rest of the load brought to rest at the same time). If coordinate values for acceleration and torque are expressed in terms of the initial angular velocity of the cam (ω_c), the stopping time (Δt), and the inertia of the cam and load (I_{cL}), the peak torque is then expressed as:

$$\text{Max torque} = \frac{I_{cL}\omega_c}{\Delta t(0.63)}$$

where 0.63 is still the ratio between average and peak torque for one quarter of a sinusoidal curve. This shows that, in general, the magnitude of collision forces is enlarged by increases in:

1. The inertia of colliding bodies
2. The relative velocity of colliding bodies
3. The stiffness of colliding bodies (stopping times are shorter for stiffer bodies).

If energy is lost during the collision, impact forces are reduced. This might seem strange at first, but again, the forces are generated in an attempt by the two bodies to conform to each other's velocity upon impact; and should some of their total kinetic energy be lost in the process, they can conform more readily. For this reason, dampers; bumpers; dashpots; friction; etc., can often make intermittent motion devices run more smoothly.

The quasi-static model does not automatically allow us to find exact values of collision times and forces then, but it does help us improve our approximations and our understanding.

The Quasi-Static Model and High Speed

The quasi-static model of an elastic body has been used to study the torque produced during the collision of two machine members; an impact situation. This model, however, is not limited to design problems involving impacts. It can be used to analyze Genevas or cams as readily as escapements or ratchets. However, it is only necessary to use it when the forces generated in a machine are large enough to cause elastic deformation in the various links and structure of the machine. This happens at surprisingly low velocities in certain types of intermittent motion mechanisms such as ratchets, interrupted gears, escapements, etc., and at much higher velocities in all mechanisms, including the best cam systems.

Spring Rates for Machine Linkages

A more accurate determination of force and stress levels in intermittent motion mechanisms might be had if an estimate could be made of the relationship between force level and deformation of the body surfaces during a collision; or in other words, if it were possible to estimate the rate of the spring assumed to exist between the colliding masses in the quasi-static model.

Some experimenters have found that it is possible to determine a force-deformation relationship through experiments with the actual bodies in so-called static testing apparatus. Such apparatus has to be capable of generating very high forces and measuring very small motions, but it does exist. Spring rates determined this way can be used to estimate some impact forces. In very high-speed machinery, however, spring rates obtained by these methods do not appear to be useful. Bodies appear to be significantly stiffer at high-impact velocities than they do at low-impact velocities. Everyone must have heard of straws that have been imbedded in trees by tornadoes. At low velocities the straw would crumble long before it would penetrate a piece of wood, but at extremely high velocities such penetration does, in fact, occur. We do not, of course, deal with velocities of this sort in most machine design situations, but those working on the frontier of machine speed and performance may have difficulties should they try to predict exact force levels using spring constants determined statically.

VIBRATION

Vibration of Elastic Bodies

We cannot deal intelligently with elastic bodies without having some feeling for the principles of vibration. Just as everything concerned with elastic

body mechanics, this is an extremely complex subject. However, I feel that an understanding of a few basic principles will be sufficient to provide a better feel for the problems and performance of intermittent motion mechanisms. The principles of this section cannot be used to determine vibration frequencies or amplitudes of a specific design situation, but hopefully, enough information is included to allow for an approximate guess about what is going on and what can be done to improve a design if improvement is required.

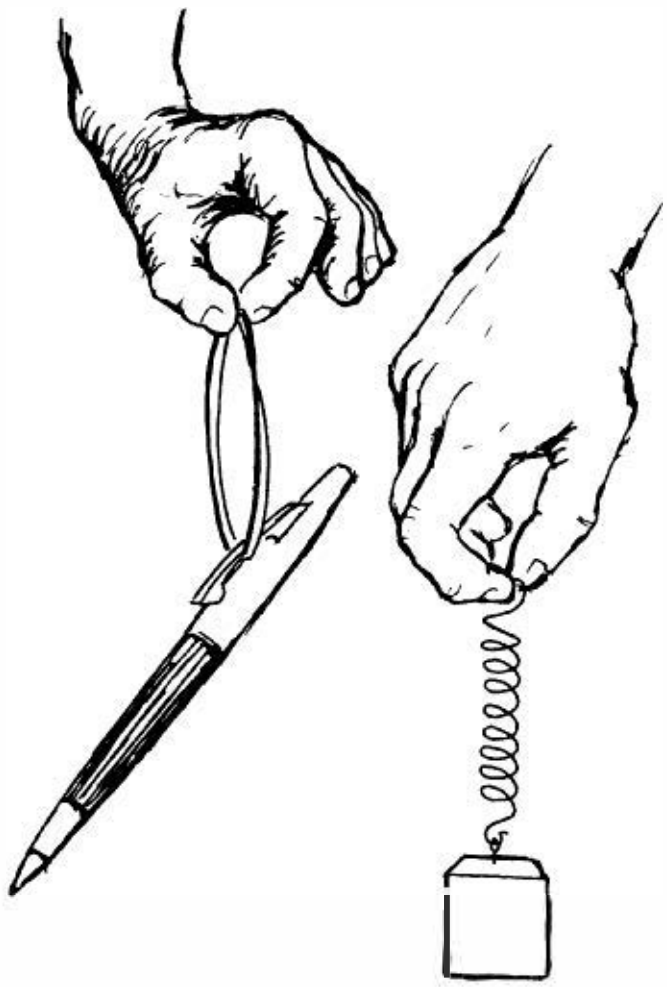


Fig. 3-6. Hang a pen from an elastic band to study the motion of a mass-on-a-spring as you read the section on vibration. Use a lightweight elastic band.

The Block on the Spring—Forced Undamped Vibration

To begin, the motion of a block that is hanging from a spring should be studied. The upper end of the spring will be moved up and down at various speeds, thus affecting the block. To help visualize the situation, I suggest you assemble a pencil and elastic band (or a block on a spring) as shown in Fig. 3-6. The elastic band should be a fairly soft one.

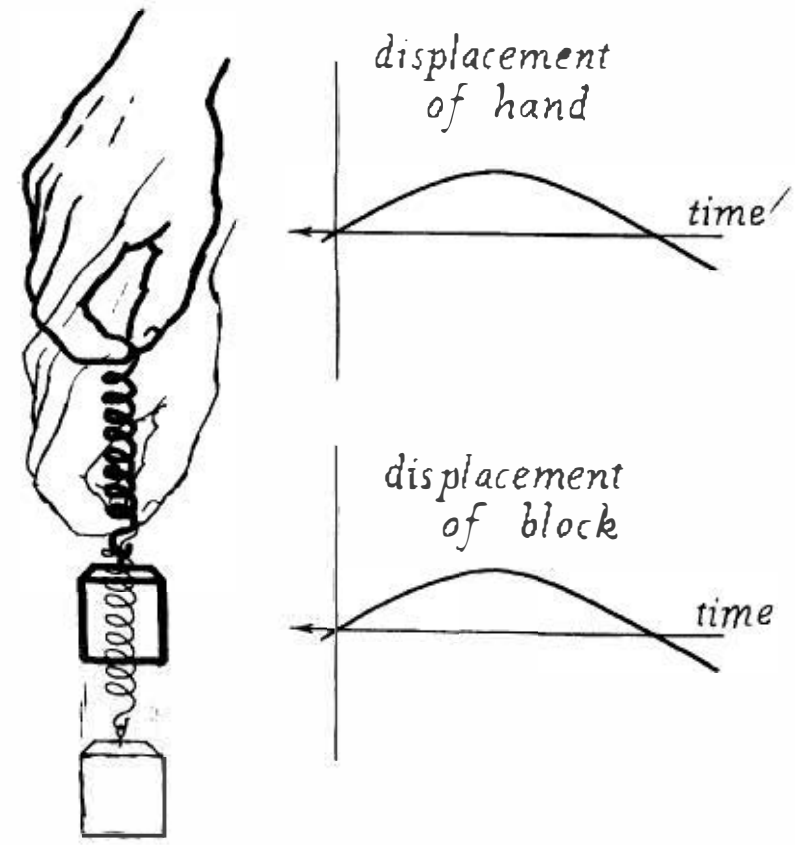


Fig. 3-7. Motion of a mass-spring with very slow input displacement (exciting frequency much lower than the resonant frequency).

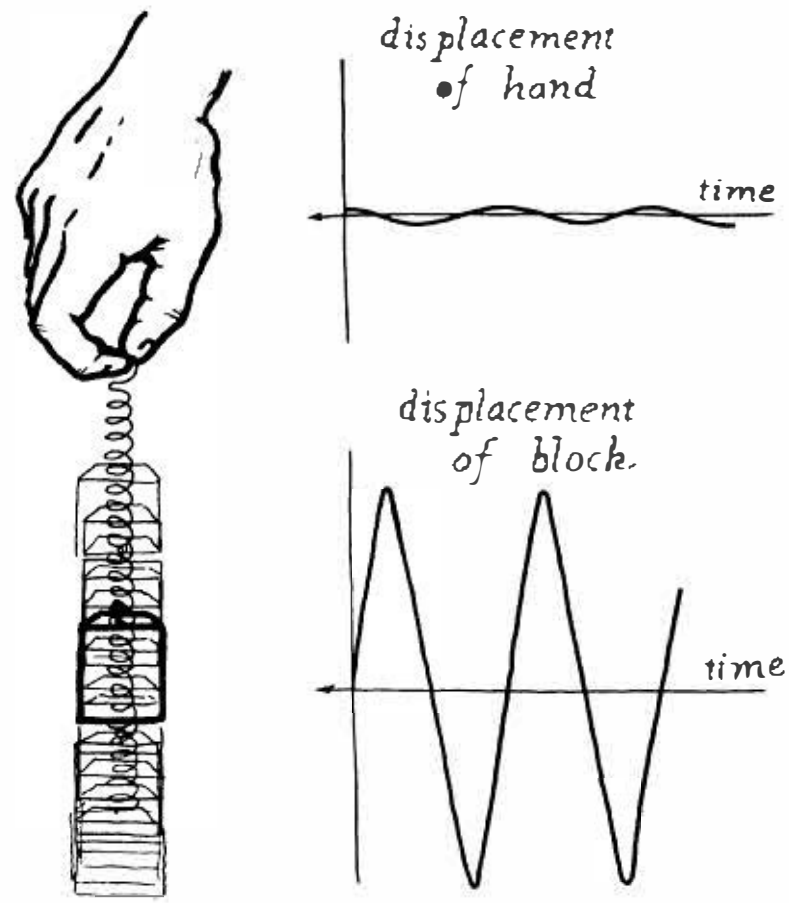


Fig. 3-8. Motion of a mass-spring at the resonant frequency.

If you move your hand up and down very slowly, the block will follow the motion of the hand exactly, moving up and down by the same amount (amplitude) and at the same time (same frequency and same phase as in Fig. 3–7).

As you start to move your hand up and down more rapidly (higher frequency), you will notice that the block begins to move through a wider and wider displacement. Eventually, a frequency of input will be reached that causes the block to go through an exceedingly large amplitude of motion; and it will then take only a very little motion of the hand to keep the block going. This is called the resonant frequency of the spring mass system (block on spring) that is being shaken. The block and hand are still moving with the same frequency but the amplitude of motion of the block is much greater than the amplitude of motion of the hand (Fig. 3–8). There is also a phase change. It is hard to detect, but the block is lagging the hand by one-quarter "wavelength" of the input frequency.

If you continue to move your hand more rapidly (still higher frequency) you find that the motion of the block dies down and, in fact, that you can move your hand at some very high frequencies which the block just won't follow at all. The motion of the block becomes less than the motion of the hand (Fig. 3–9). The block too, incidentally, now lags the hand by one-half wavelength.

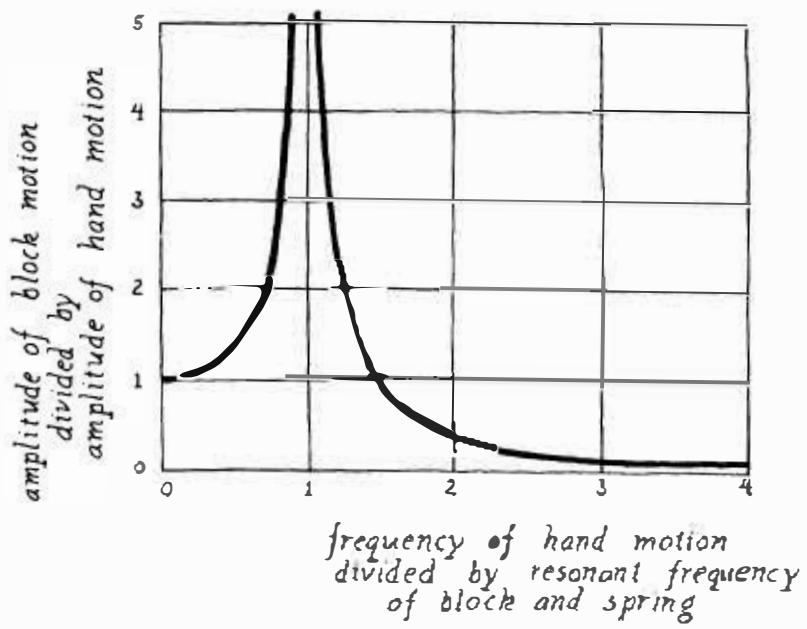


Fig. 3–10. Curve showing the motion of an undamped mass-spring as a result of various inputs.

If the ratio of the amplitude of displacement of the block over the amplitude of displacement of the hand versus a ratio of the frequency with which the hand is moved up and down divided by the resonant frequency of the spring mass system is now plotted, a curve such as that shown in Fig. 3–10 will result. This curve illustrates the forced, undamped vibration of a simple spring-mass combination (forced, because in each case the hand kept the block moving at a frequency determined by the hand). All this should be tried; it is a very informative experiment for people who want to understand vibration and it only takes a few minutes to do.

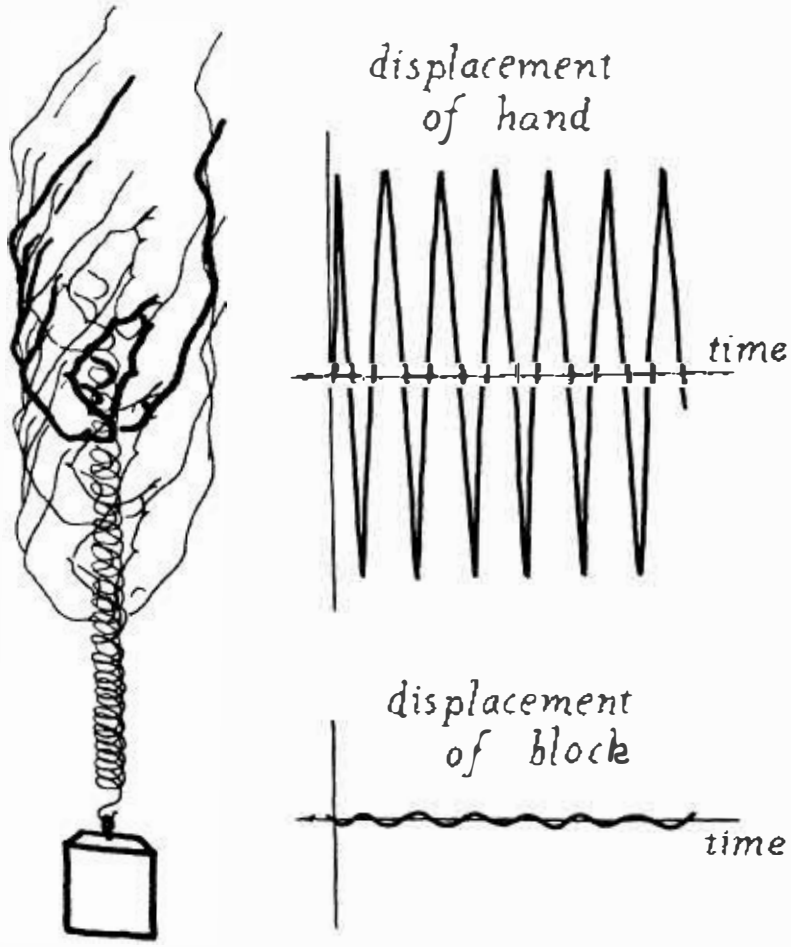


Fig. 3–9. Motion of a mass-spring with very rapid input motion (exciting frequency much greater than the resonant frequency).

Machine Vibrations

At first glance, the motion of the block seems to violate our old friend $F = ma$, especially at the resonant frequency point. The displacement of the hand is small and constant, yet the motion of the block becomes more and more agitated. If the force exerted on the spring by the hand could be measured, however, it would be found that this force gets stronger as the amplitude of displacement of the block increases. If graphs could be derived for acceleration, velocity, and displacement, as before, they would show that the displacement does, indeed, become increasingly large as time progresses. This

same phenomenon happens in a machine if it is driven at a resonant frequency. A small, repetitive input can produce increasing displacement, distortion, forces, and stresses in the machine until, as in some instances, the machine destroys itself.

The resonant frequency of a spring-mass system is proportional to the square root of the spring rate divided by the block's mass, or:

$$f_R = \frac{1}{2\pi}\sqrt{\frac{K}{m}}$$

Most machine members have a very high spring rate, as has already been seen. They also have a significant mass, but the spring rate is so high that the resonant frequency of a normal machine linkage is extremely high (thousands of cycles per second is probably a typical number). Nevertheless, in very high-speed systems, resonant problems can indeed be encountered; and, of course, in machines involving actual springs with relatively slow rates, resonant problems can develop at quite low frequencies.

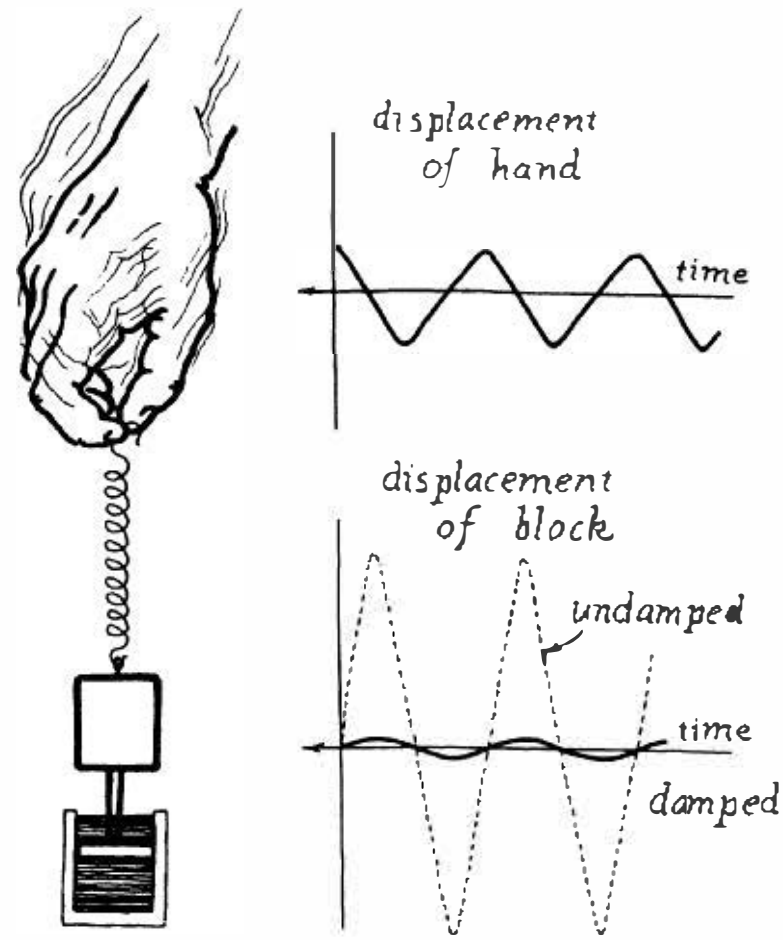


Fig. 3-12. Motion of highly damped spring-mass at the resonant frequency.

Forced Damped Vibration of a Mass-Spring Combination

The vibrating springmass system described is essentially an "undamped" system; meaning a system in which there are no energy losses. There are, of course, slight energy losses in the spring as well as other losses caused by the friction of the air on the block, but there is not enough damping to alter significantly the motion of the block. In most machine design situations, there is quite a bit of friction present, not only in spring losses and air friction, but in bearing friction, slide friction, internal energy losses, etc. And in many designs devices called "dampers" are introduced, to produce energy loss and alter the performance of the machine. Figure 3-11 shows what fluid or viscous damping will do to a resonant spring-mass combination. Again, it is assumed that the hand is moving up and down to shake a small block mounted on the end of a spring, but in this case, the block is fastened to the piston of an oil-filled dashpot.

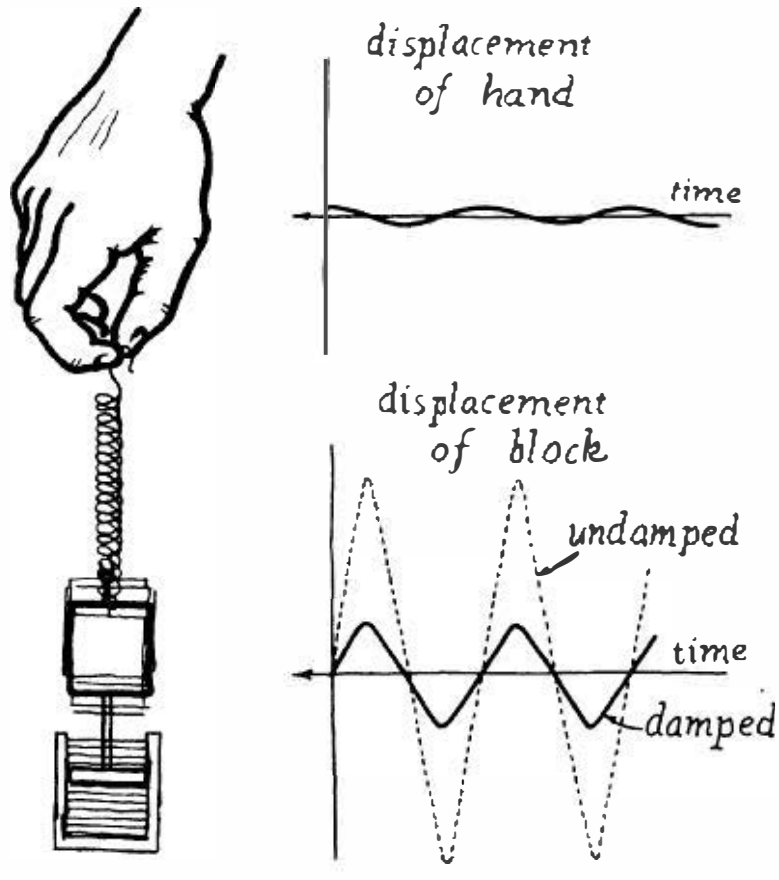


Fig. 3-11. Motion of a lightly damped spring-mass at the resonant frequency.

Figure 3-12 shows another damped block. This time, the viscosity of the damping fluid has been increased. It now takes quite a bit of motion of the hand, even at the resonant frequency of the block-spring, to cause detectable motion in the block. Note that in every case the frequency of oscillation is the same as the forcing frequency (frequency of the

hand motion). Only amplitude and phase are changed by damping and/or the relationship between resonant frequency and forcing frequency.

Dampers

We have seen how the motion of the block is suppressed by fastening it to a dashpot. Experience proves that this will happen, but we would probably be in trouble if we had to specify a certain amount of damping for a particular design situation. What do vibration experts mean by *damping coefficient* or *damping constant* and how could one of these be measured in an actual design situation?

A damper is a device that acts to retard the motion of the mechanism to which it is attached, by producing a force which opposes this motion. There are two kinds of dampers that work by dissipating energy:

1. *Frictional damping:* Friction forces, of course, oppose the motion of moving parts. This force is generally independent of the relative velocity of the parts involved—it is constant once motion has started. It is dependent on contact forces, surface finish, lubrication, and the properties of the materials involved. Friction is not normally used to damp vibrations because it almost invariably leads to wear, but damping is often a side benefit of that friction we have been unable to eliminate from our designs. Friction damping is also called *coulomb damping.* This kind of damping is always the dominant form of damping at low velocities (as moving bodies approach the rest condition) and low stress levels.
2. *Viscous damping:* A true viscous damper produces a force which is proportional to the relative velocity of the moving parts and the damper. You can see the effect for yourself if you try to move a paddle through oil or water. If you move the paddle slowly there is little resistance to motion. Try to move it rapidly, however, and you encounter a sudden increase in resistance (generally proportional to the square of the velocity of the paddle). It is this property of the viscous damper, its dependence on velocity, which makes it so useful as a vibration damper.

Viscous dampers are rated in terms of a damping constant which is determined by the following expression:

$$F = Cv$$

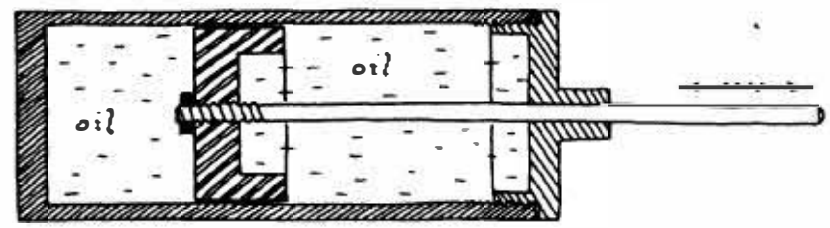


Fig. 3-13. Dashpot damper; resists linear motion in the direction shown by the broken arrow. The piston must be a loose fit in the cylinder.

where: F = force; lbs

C = damping constant: lb-sec/ft

v = velocity; ft/sec

This will be better understood by looking at an example.

Dashpot

A dashpot, in its simplest form, is an oil-filled cylinder in which moves a sliding shaft and a piston. The outer end of the shaft is connected to the machine whose motion is to be damped. The faster one tries to move the shaft, the greater its resistance to motion. (See Fig. 3-13.)

If it takes two pounds of force to move the shaft six inches in 20 seconds, the damping constant is 80 lb-sec/ft, and is found as follows:

$$\text{Velocity} = \frac{\text{Distance}}{\text{Time}} = \frac{0.5 \text{ ft}}{20 \text{ secs}} = 0.025 \text{ ft/sec}$$

$$C = \frac{F}{v} = \frac{2}{0.025} = 80 \text{ lb-secs/ft}$$

To change the damping constant of the dashpot, increase or decrease the viscosity of the oil, or the size or configuration of the piston. Adding holes in the piston, for example (so that the oil can pass from one side to another more easily as it moves), will decrease the resistance of the piston to motion.

Rotary dashpots are also widely used to reduce vibrations. Damping will be discussed further in subsequent chapters, but we now have nearly enough information to finish the discussion of basic vibration.

Critical Damping

For every spring-mass system there is some degree of damping that is considered "critical." As will be seen in a later section, if a single shock or blow is applied to a spring-mass system it will usually oscillate for awhile, but the oscillations will gradually

die away because some sort of damping is always present. As the damping is increased, the oscillations die away more rapidly. In fact, above some level of damping, the mass will not oscillate at all, but will merely go through a portion of one cycle of motion before returning to its initial position. (See Fig. 3–17.) Critical damping is that degree of damping which causes the mass to return to rest in the shortest possible time without oscillation. The critical damping coefficient, C_c can be found by applying the equation:

$$C_c = 2\sqrt{Km}$$

where: K = spring rate of the spring; lbs/ft
m = mass; slugs

Note that "damping coefficient" only refers to viscous dampers; it is not used to define coulomb or internal damping.

Forced Vibrations—Summary

Now a set of curves can be drawn that summarizes the performance of the block and spring under various forcing frequencies and damping conditions. (See Fig. 3–14.) Each curve is for a different damping factor, which is defined as:

$$\text{Damping factor} = \frac{\text{Damping constant of actual damper used with spring-mass}}{\text{Critical damping coefficient for same spring-mass}}$$

As the stiffness of the dashpot or other damper used is increased, a lower curve to predict performance of

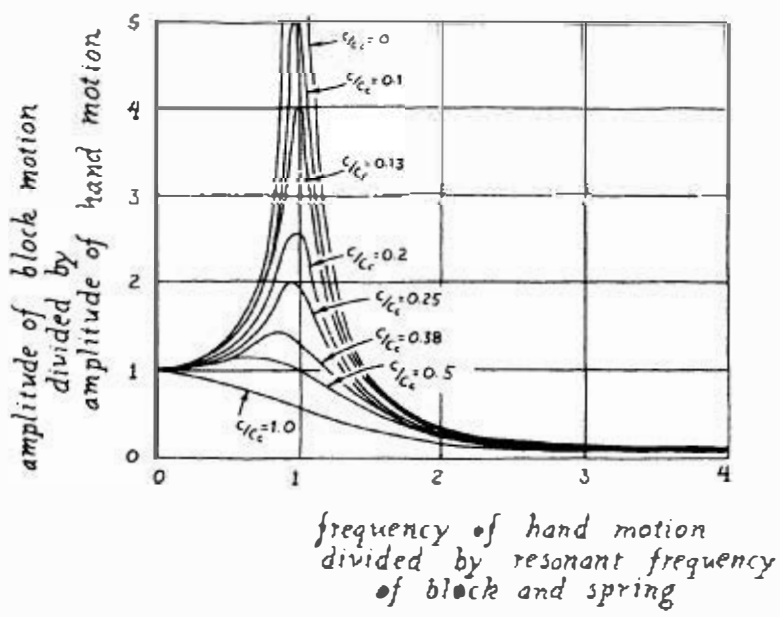


Fig. 3–14. Complete set of motion curves for damped and undamped spring-mass systems.

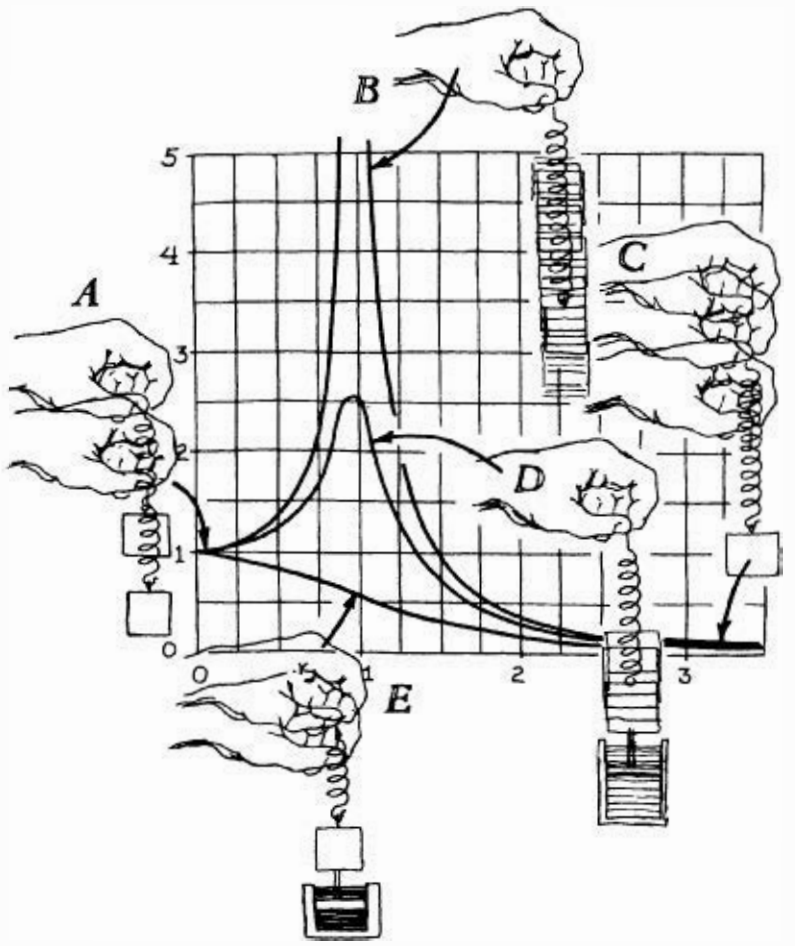


Fig. 3–15. Relationship between motion curves and the various mass-spring experiments of Figs. 3–7 through 3–12—(A) Fig. 3–7: Input frequency much lower than the resonant frequency; (B) Fig 3–8: Undamped motion at the resonant frequency; (C) Fig. 3–9: Motion at high-input frequency; (D) Fig. 3–11: Light damping at the resonant frequency; (E) Fig. 3–12: Heavy damping at the resonant frequency.

the spring-mass would have to be used. Again, damping factor, like critical damping, is used only to describe viscous dampers.

Figure 3–15 shows the same set of curves as Fig. 3–14, but the block-springhand drawings have been superimposed on the curves to relate the previous experiment to the curves. Note that damping is really effective only near the resonant frequency of the spring–mass. At higher and lower frequencies the curves all run together, regardless of degree of damping.

Free Vibrations—Shock Excitation

In all of the cases considered so far, we have dealt with a steady or repetitive input. This is called *forced vibration*. The hand continuously moved up and down while the motion of the block was studied. In many machine situations, especially those involving intermittent motion mechanisms, the inputs are only momentary. A machine link is subjected to a sudden shock, for example. Again, using our hand-spring

mass-dashpot system to illustrate the situation, Fig. 3–16 shows what might happen. The hand is moved up and down once, then stops. The block again vibrates sinusoidally, but this time with a decreasing amplitude. The frequency of vibration in this case depends upon the degree of damping present. If damping is light, then the frequency of the block is found by the expression:

$$f = f_N\sqrt{1 - c_f^2}$$

where: f = damped frequency of vibration of the block-spring
f_N = undamped resonant frequency of the same block-spring
c_f = damping factor (ratio of actual to critical damping).

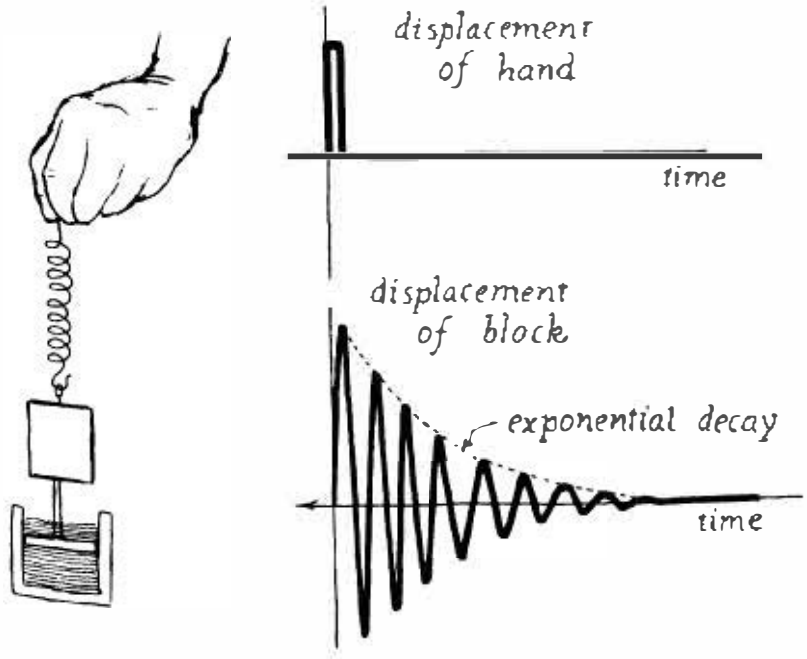


Fig. 3–16. Motion of spring-mass to single-shock input with light viscous damping.

With critical or larger than critical damping there will be no oscillation at all. The block will return to its original position with what is known as aperiodic motion. (See Fig. 3–17.)

If damping is light, the rate at which the motion of the block dies away is determined by the amount of damping present. If energy loss occurs only inside the spring and through air friction, the block might remain in motion for several seconds. If the motion is suppressed by a viscous damper, however, its motion will cease very rapidly, the displacement decreasing exponentially with time, as shown in Fig. 3–16. If a friction damper is used, as in Fig. 3–18, the decay is linear and the frequency of vibration is

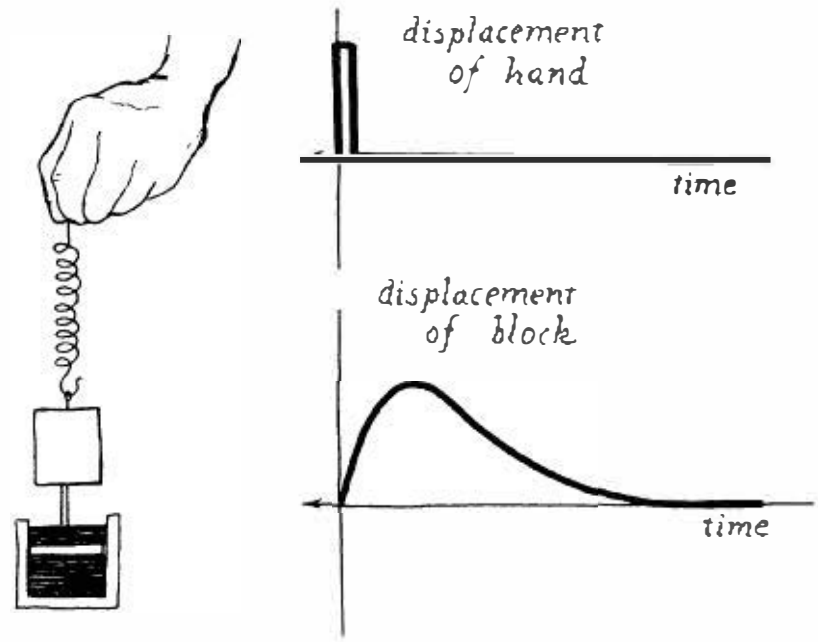


Fig. 3–17. Motion of spring-mass to single-shock input with critical viscous damping. The response to heavy viscous damping is similar to this, but the block will take longer to return to its original position.

equal to the undamped resonant frequency of the block, regardless of the amount of damping present.

The amplitude of motion of the block will depend upon the exact motion of the hand that has excited it. If the hand moves through a cycle whose duration, or *period* as it is called, is equal to the duration of one cycle of the block and spring at resonant frequency, the block's motion will be more extreme than if the hand moves slowly or very rapidly, with respect to the resonant frequency. The response of the block to a "square wave" motion of the hand

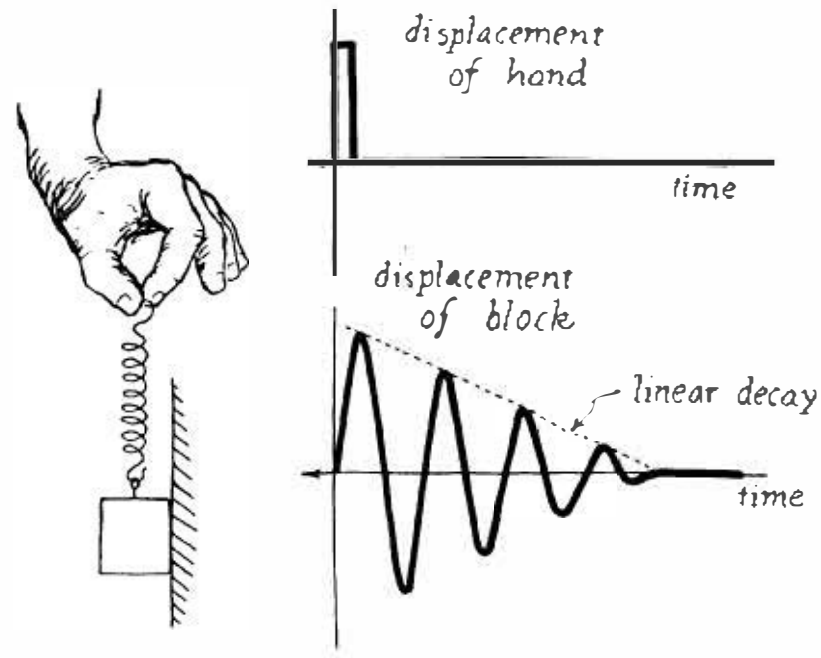


Fig. 3–18. Motion of spring-mass to single-shock input with friction damping. Compare with Fig. 3–16.

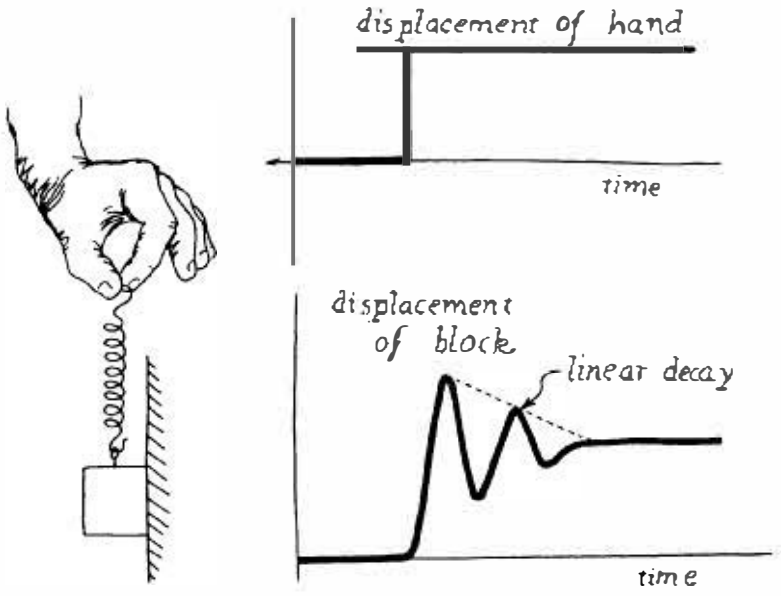


Fig. 3–19. Motion of a spring-mass to a single-step input with friction damping. Compare with Fig. 3–18.

will be different from that to a "sinusoidal" motion of the hand, although in all cases, the general motion of the block will be a sinusoidal motion decreasing in amplitude, with time.

The maximum acceleration in the block for single inputs of this kind is usually about two times the acceleration of the input. The acceleration can be much greater than twice, for repeated inputs, if they occur at a frequency which tends to excite resonant motion in the block.

Free Vibrations—Step Excitation

A slightly different situation is shown in Fig. 3–19. The hand moves upward once but remains in the higher position. The block moves upward also, moving farther than the hand, then oscillating, with the oscillations again dying away. The block will finally come to rest with a change in displacement equal to the change in displacement of the hand. The illustration shows this situation with coulomb damping. With viscous damping the oscillations would decrease exponentially. In fact, everything said about free vibrations also would apply to this new situation; only the final displacement is different.

This situation will again be encountered when we discuss stepping motors. Figure 14-31, for example, shows the displacement-versus-time of a stepper that is damped. It is the same "curve" as that in Fig. 3–19.

Coupled Oscillators

There are not many one-link machines, so the single block on a single spring, while helpful, must be expanded if we are to understand the vibrations of actual designs. In Fig. 3–20, therefore, two blocks are connected by three springs to each other and to the ground. This is a good quasi-static "model" of a two-link, elastic body mechanism.

If the top block (M_1) is moved up and let go, oscillations will be produced in the blocks as shown in Fig. 3–20. Block M_2, for example, will vibrate with an amplitude that keeps changing, getting periodically larger and smaller as shown in the sketch. Its frequency of vibration will depend upon the rates of the three springs in the system. The frequency with which its amplitude increases and decreases will depend upon the various resonant frequencies of the spring-mass combinations in the system. This change in amplitude frequency, incidentally, is called the "beat frequency" of the block M_2, and is a function of the difference between the resonant frequency of one of the oscillators in the system and the resonant frequency of the others. As shown in Fig. 3–20, block M_1 will vibrate in a fashion similar to M_2, but out of phase with M_2. In general, the motion of coupled oscillators can be found by graphically adding the motions each spring-mass pair would have as a separate isolated oscillator.

If this were a machine, and the hand that moved M_1 were the input drive crank, we can see that it might have some difficulty in driving block M_2; at least the motion of M_2 appears to bear very little

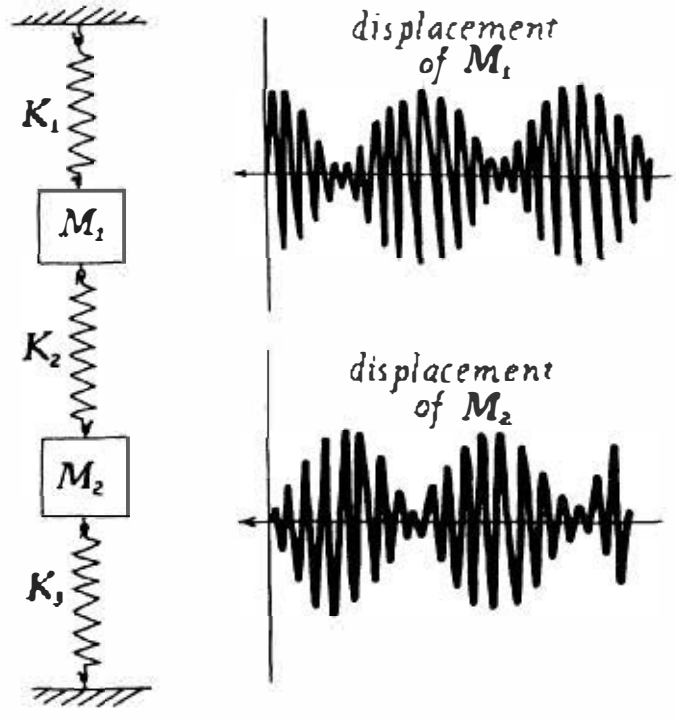


Fig. 3–20. Motion of coupled oscillators following single-shock displacement of block M_1. If damping were present (it always is, in practice) these oscillations would die away rapidly following a pattern similar to Figs. 3–16, 3–17, or 3–18.

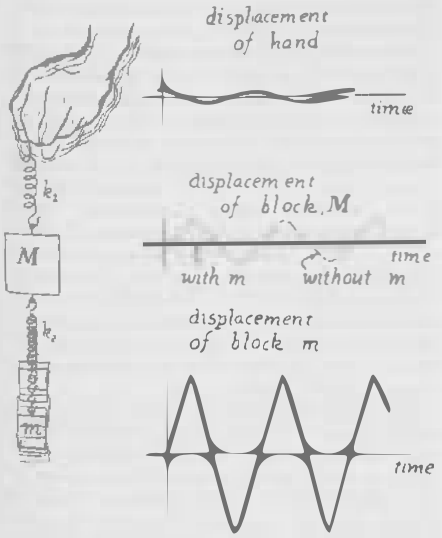


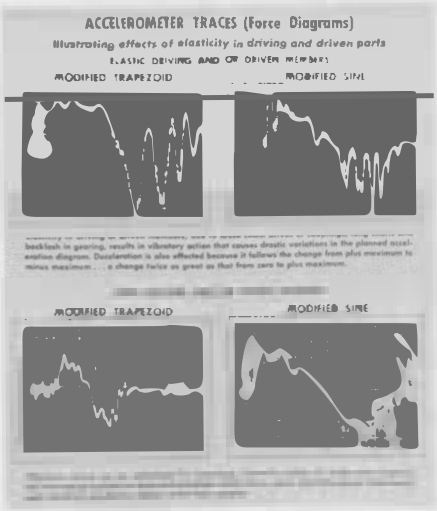
Fig. 3–21. Using a properly tuned coupled oscillator (the lower spring-mass) to damp the motion of the main spring-mass at one frequency, the resonant frequency of the smaller spring-mass.

relationship to the motion of the input; which may be the motion we desire to transmit through the machine. In an actual design, the "springs" are so stiff that the motion of M_2 probably follows that of the driver much more closely than this, however. In fact, the resonant frequencies of the various links in a mechanism are so high that most input motions represent a very small frequency with respect to the natural frequency of any of the oscillators involved, and the output motion will follow the input on nearly a one-to-one basis. As we increase the speed of the machine, however, the drive frequency more nearly equals the resonant frequency of some links, and we can reach a point at which the output link will gallop at a beat frequency, such as is illustrated by this two-block experiment.

In most machine designs the masses and spring rates (M_1, M_2 and K_1, K_2, K_3) shown in Fig. 3–20 will not be equal to each other. If K_1 and K_2 are very stiff compared to K_3, then M_2 will follow the motion of M_1 fairly closely. If K_2 is very large, compared to K_1 and K_3, we can even assume that K_2 does not exist as a spring, and that we have a mass equal to $M_1 + M_2$ hanging between springs K_1 and K_3. This will affect the resonant frequency of the system (the motion of the blocks), and they will move together.

If K_2, on the other hand, is very soft compared to K_1 [illegible] almost independently. In practice, many different patterns of motion are possible, and the larger the number of mass-spring combinations involved, the greater the number of different patterns that can be found.

An interesting two-body situation is shown in Fig. 3–21. The large block M, will have zero displacement, even with a forced input from the hand, if the input frequency equals the resonant frequency of the lower block and spring (m and k_2). The lower block-spring is called a vibration absorber and it exerts an equal and opposite force on M while the hand is trying to move M. Unfortunately, this type of vibration control is only good at one frequency, the resonant frequency of m and k_2. M and k_1 now have two resonant frequency points, one above, and one below, the resonant frequency of M and k_1 alone, thus this trick can only be used in a constant speed system.



Photographs courtesy of the Ferguson Machine Company

Fig. 3–22 Vibration patterns obtained from high-speed machinery, showing that the pe[illegible]nd has its counterpart in machine design.

We have used a pen, hanging from an elastic band, to study vibration. The principles revealed by these experiments, however, definitely apply to the practical world of machine design, as is proven by the acceleration curves shown in Fig. 3–22. Here we see the smooth acceleration curves produced by high quality cams (see Fig. 8–5, *Left*, for example) modified by the elasticity of the machine. The pen-on-a-rubber-band should help the designer understand the changes that must be made in a machine to reduce the amplitude of vibration to a tolerable level.

IMPACT

Impact and Intermittent Motion

A really good understanding of elastic body mechanics can never be achieved until we understand impact or what happens during a collision between machine members. Throughout this book the attempt has been made to determine collision forces, duration times, and impact effects by using primarily graphical methods. A real understanding of impact, however, also involves such things as the simultaneous handling of the conservation of momentum and the conservation of energy; all very difficult to do graphically. Probably, it is not really necessary to describe these factors for a working understanding of intermittent motion devices, but some readers may prefer to have them explained. Therefore, for those who wish to know more, the following discussion on impact has been included. Although the treatment is mathematical rather than graphical, the mathematics involved has been kept to a minimum and very likely, most readers will be able to follow the discussion. The article is mainly verbal and tries to show exactly what happens during impacts, why exact "answers" are difficult to obtain, and how to maximize or minimize the effects of impacts to solve machine design problems.

Basic Concepts

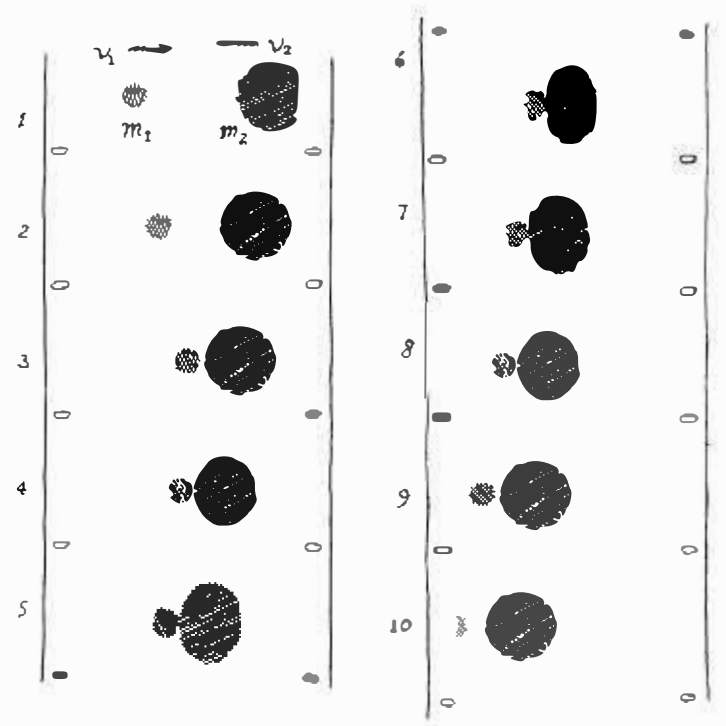
Important properties involved in the study of impacting bodies include:

Velocity v_1 (Body 1); v_2 (Body 2)
Mass m_1, m_2
Momentum (m_1v_1), (m_2v_2)
Kinetic energy $(m_1v_1^2/2)$, $(m_2v_2^2/2)$

Velocity and momentum are vector quantities and must have proper signs. Conventionally, travel toward the right is assigned a positive value; travel toward the left, a negative value.

Sequential steps in a collision between two elastic spheres are shown in Fig. 3–23. In the first two frames, the bodies are approaching each other, each aimed at the center of gravity of the other. In the next few frames, the bodies meet and start to push each other. Each deforms under the (equal and opposite) forces exerted by the other. During this period, part of the original kinetic energy of each body is converted to potential or "spring" energy within the two bodies, and part is converted to heat within the bodies. The remainder of the original energy is neither lost nor converted; the two bodies remain in motion during the entire impact process. By the fifth frame, the compression phase of the impact is over. The bodies are now traveling with a common velocity toward the left.

In the next three frames, the bodies return to their original shapes (if they are sufficiently elastic). Some of the potential energy they have stored is reconverted to kinetic energy. Some additional energy is lost as heat. This is called the "extension" phase of the impact. In the final frame, the bodies are again traveling their separate ways. Their individual masses have not changed, but each has a new velocity, a new momentum, and a new kinetic energy. Part of the original system energy has been lost, and the rest has been redistributed between bodies.



Drawings courtesy of MACHINE DESIGN Magazine; May 9, 1968; pp. 165–178

Fig. 3–23. Simulated motion picture of an impact between two spheres.

The velocity of each body, following the collision, can be determined by solving the simultaneous equations of momentum and energy. These equations are given below, in their general form; the signs used for each velocity term would have to be chosen carefully to reflect a specific situation, such as the collision illustrated in Fig. 3-23. If you follow the sign conventions suggested in Chapter 2 you should have no problem.

1. The change in momentum in Body 1 equals the change in momentum in Body 2:

$$m_1v_1 - m_1v_1' = m_2v_2' - m_2v_2 \qquad (3\text{–}1)$$

where v_1' and v_2' are the respective velocities after impact. Note that in this case v_1 is positive but v_1', v_2 and v_2' are negative.

2. Total system energy before impact equals total system energy after impact plus energy lost during impact (Q):

$$\frac{m_1v_1^2}{2} + \frac{m_2v_2^2}{2} = \frac{m_1(v_1')^2}{2} + \frac{m_2(v_2')^2}{2} + Q \qquad (3\text{–}2)$$

These equations can be solved only if the amount of energy lost during the impact can be determined. It is common to express the lost energy as a function of the relative velocities of Body 1 with respect to Body 2, before and after impact, since these velocities can sometimes be measured experimentally. The ratio $(v_2' - v_1')/(v_1 - v_2)$ is called the coefficient of restitution, K. Energy loss, Q, can be solved from:

$$Q = \tfrac{1}{2}(1 - K^2)\frac{m_1m_2}{m_1 + m_2}(v_1 - v_2)^2 \qquad (3\text{–}3)$$

In a perfectly elastic collision (no energy loss), the coefficient of restitution equals unity; in a perfectly inelastic collision, K equals zero (all energy converted to body deformation is lost). All system energy is not lost in the latter case, however, since the two bodies can still be in motion after the collision.

Other commonly used equations give the velocity of each body following the collision and can be derived by algebraic manipulations of Equations 3–1, 3–2, and 3–3:

$$v_1' = \frac{m_1v_1 + m_2v_2 - Km_2(v_1 - v_2)}{m_1 + m_2} \qquad (3\text{–}4)$$

$$v_2' = \frac{m_1v_1 + m_2v_2 + Km_1(v_1 - v_2)}{m_1 + m_2} \qquad (3\text{–}5)$$

And the expression for the energy the system must lose (in single or multiple impacts) before the colliding bodies will move together as one, with a common velocity, is:

$$Q = \frac{1}{2}\frac{m_1m_2}{m_1 + m_2}(v_1 - v_2)^2 \qquad (3\text{–}3a)$$

Determining Energy Loss

Most textbooks imply that the coefficient of restitution and, therefore, impact energy loss, is merely a function of the materials involved in the collision. True, lead and most plastics, for example, convert more impact energy to heat and distortion than do elastic materials such as hard steel or ivory. But materials are only one of many factors that determine energy loss in an actual collision between machine members.

The shape of colliding bodies also affects the amount of energy lost during an impact. Other things being equal, a block will not bounce as high as a ball. Particularly important is the shape of the actual contact area. If the block lands on a corner and the corner deforms, the energy loss is increased further. (See Fig. 3–24.)

Impact velocity also has an effect on the coefficient of restitution and energy loss. At typical machine velocities, the apparent coefficient of restitution decreases only slightly (slight increase in energy loss) as impact velocity is increased. At higher velocities, even elastic bodies deform permanently on impact, and a larger percentage of energy is lost. At still

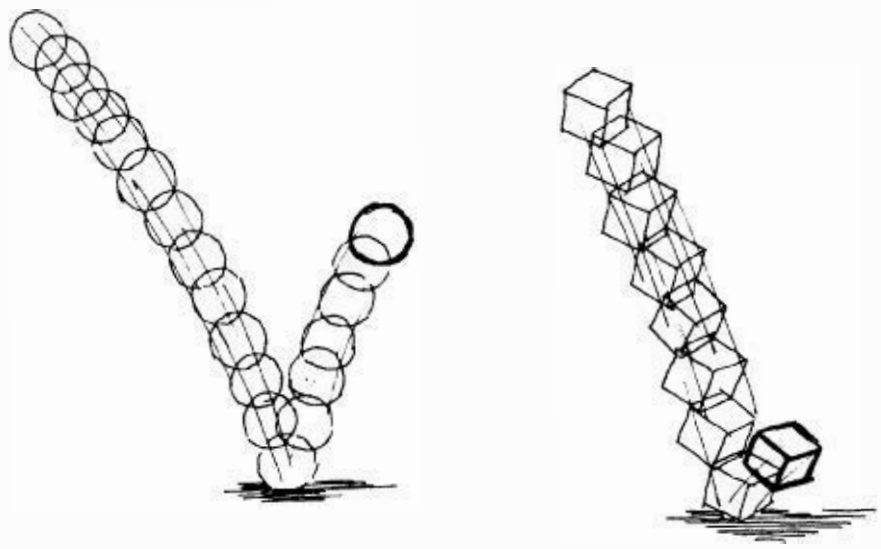


Fig. 3–24. The coefficient of restitution depends on the shapes of colliding bodies as well as on their materials. A sphere will bound higher than a block, for example.

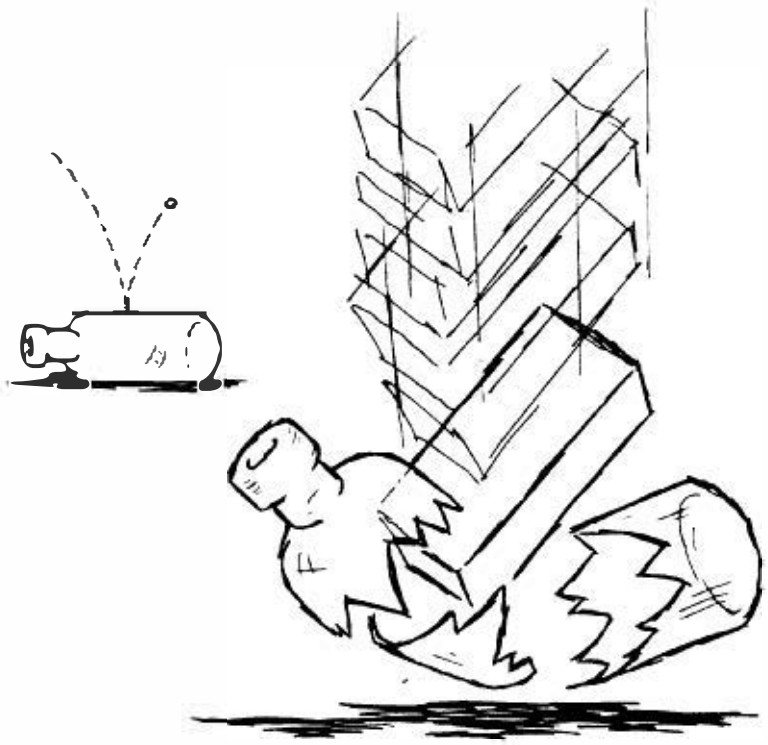
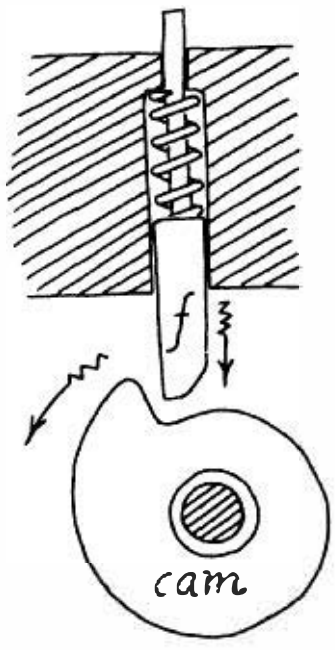


Fig. 3–25. The coefficient of restitution also depends upon collision velocities, or the amount of kinetic energy involved in the collision. Glass has a very high coefficient if only light impacts are involved.

higher impact energies, one or more of the bodies may fracture. (See Fig. 3–25.)

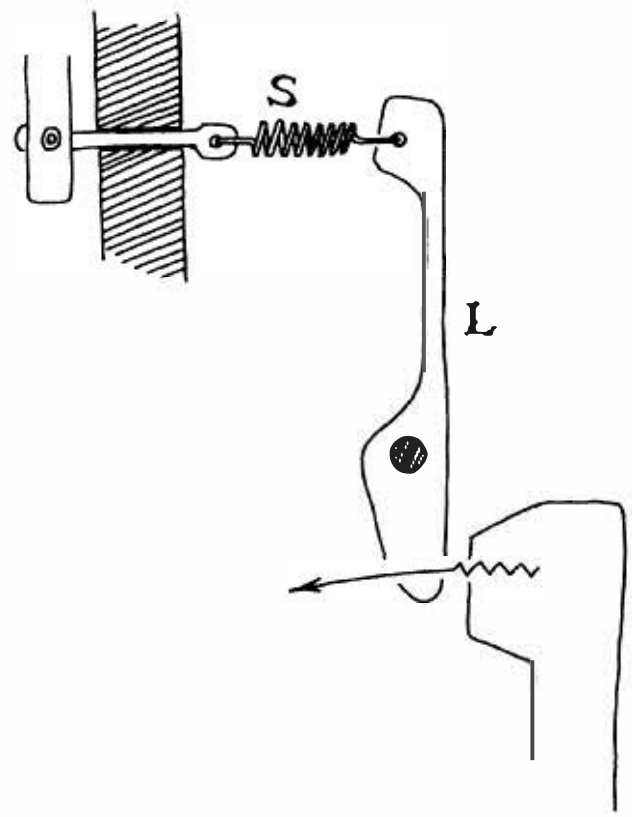
Glass is a very elastic material at low impact velocities. Its behavior at higher velocities is well known. Under certain conditions, a high-velocity shell punches a hole in heavy armor plate that looks like the hole a small bullet makes in a windowpane. This is the principle of armor-piercing shells. At lower velocities, the shell only dents the armor (plastic deformation); and at still lower velocities, it bounces off.

Sliding friction can cause the loss of a considerable amount of impact energy if the bodies move past each other during the collision. In the mechanism shown in Fig. 3–26, for example, energy is dissipated between the cam and its axle, between the cam and its follower, and between the follower and its frame during impact of the follower on the cam. These losses subtract from the total system kinetic energy just as effectively as do internal material losses. As far as bounce is concerned, the follower will behave as if it were made of lead instead of steel, if friction losses are great enough.



Drawing courtesy of MACHINE DESIGN Magazine; May 9, 1968; pp. 165–173

Fig. 3–26. Energy is lost as the follower (f) slides in its supporting block and on the cam. This loss appears to be a decrease in the coefficient of restitution. What is the coefficient of restitution now?



Drawing courtesy of MACHINE DESIGN Magazine; May 9, 1968; pp. 165–173

Fig. 3–27. Energy is stored in spring S, and link L, and is ultimately lost.

Vibrations caused by impact are another source of energy loss. This is one of the main reasons that body shape affects the apparent coefficient of restitution. Whether vibration energy is returned to the system as kinetic energy or is dissipated as heat within the bodies or in the frame of the machine, depends upon modes of vibration, type and locations of supports and constraints on the bodies, body con-

figurations and materials, and "timing" of the vibration cycles with respect to the motion of the bodies themselves. Exact calculations are extremely complicated. In the device shown in Fig. 3–27 major vibrations would occur at the spring and the link or vertical arm.

Consideration of the several factors that affect energy losses points to the conclusion that it would require a major effort and considerable experimental data to determine actual energy loss in collisions between real-life machine members. Only rarely can exact solutions be obtained for the basic momentum and energy equations. These equations can be used, however, to determine design directions and, therefore, to minimize or maximize the various effects of impact even without exact numerical values for such things as impact forces, times, or velocities.

The Impulse Concept

Impulse is defined as the product of a force acting on a body, and the time during which the force acts. In mathematical terms, if P is impulse:

$$P = Ft \tag{3–6}$$

if the force is constant during the collision (which it almost never is, in a machine), and

$$P = \int_{t_0}^{t} F\,dt \tag{3–7}$$

if the force varies.

Impulse also causes a change in momentum in a body:

$$\int_{t_0}^{t} F\,dt = m_1v_1 - m_1v_1' \tag{3–8}$$

Many scholars believe this equation was the original form of Newton's famous equation, $F = ma$. Equation 3–8 is also the mathematical statement of the Law of Conservation of Linear Momentum. This equation can be used to show why momentum is conserved in the collision of the two spheres in Fig. 3–23. During impact, the spheres exert equal and opposite forces on each other for equal times. Therefore, they produce equal impulses on each other. This results in equal changes in momentum in each body. What one loses, the other gains, and momentum is "conserved" in the system.

Collisions produce very high forces in a machine. Removing the momentum from a heavy hammer, for example, requires an impulse—a product of force and time. If the time is very short, the force must be very large. In Fig. 3–28A, for example, the hammer is stopped suddenly by a heavy anvil. The force is tremendous. In the plot of force-versus-time, the area under the curve is equal to the change in momentum in the hammer. (Fig. 3–28, B.)

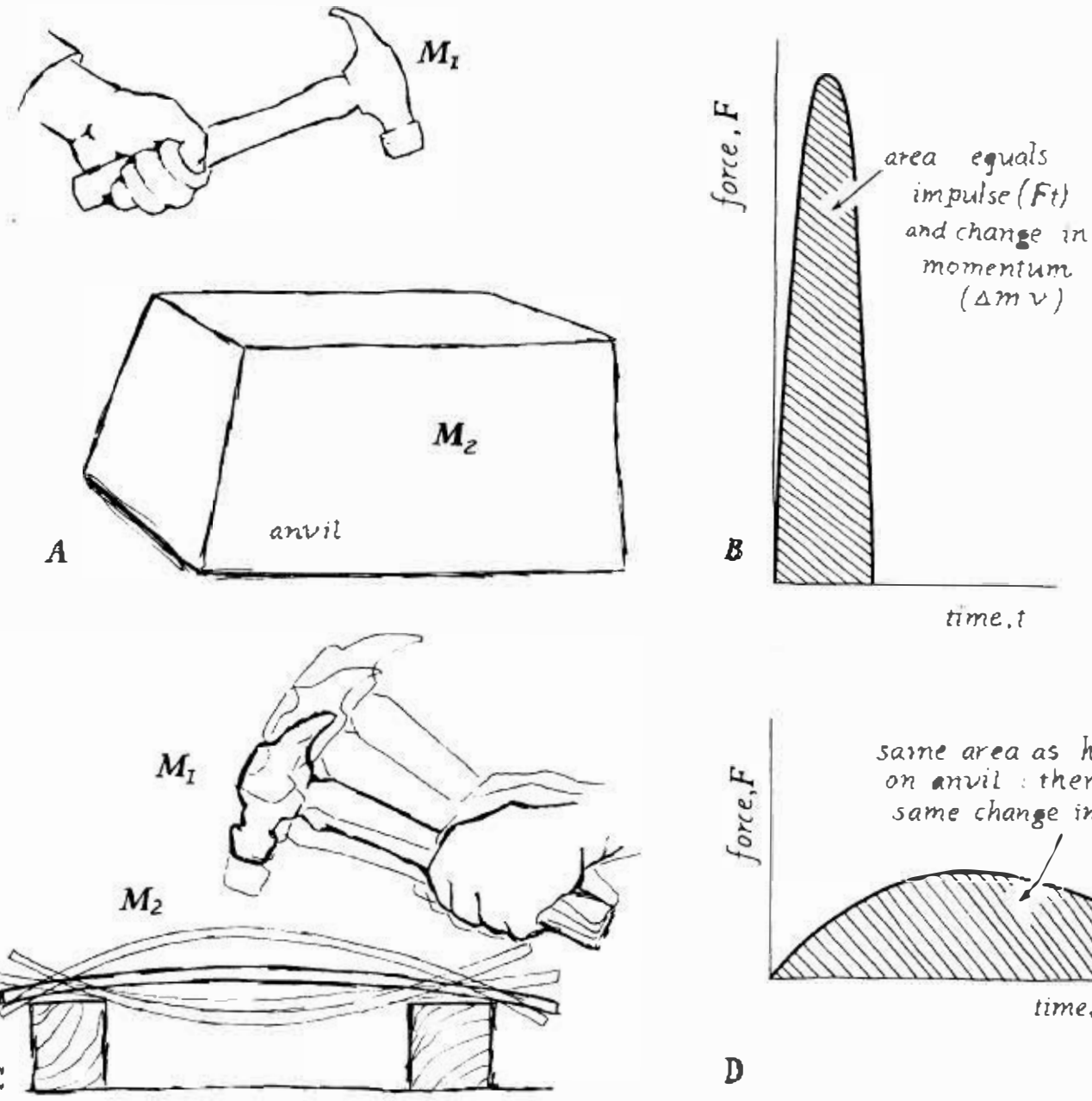
If the same hammer strikes a thin, narrow plate of steel, Fig. 3–28C, it is stopped only after a much longer time has elapsed. The blow is cushioned, and the impact forces are much lower than those produced by the anvil—even though the impulse produced by both "stops" is the same. (Areas under both curves are equal, Fig. 3–28, B and D.)

If the shape of the impulse curve or even the exact area under the curve could be determined for a given collision, velocities of the colliding bodies following a collision could be calculated without having to predict or measure the energy lost during the collision. In other words, the impulse expression (Equation 3–8), is a "substitute" for Equations 3–1 and 3–2. Unfortunately, however, it is just as difficult to predict impact forces as it is to predict impact energy losses. Impact forces during both compression and extension phases of the impact must be determined to produce a plot of force-versus-time, and these forces depend upon many factors in addition to the momentum of the colliding bodies. For example, impulse forces increase with an increase in the spring rates (force divided by displacement produced by that force) of the bodies involved in the impact.

Unfortunately, spring rate is a function of many of the things that affect energy loss—body materials, body shapes, impact energy, and velocity levels—and thus, cannot be predicted accurately. Static tests to measure spring rate have been found to be particularly misleading. In any event, spring rate might be considered a measure of the rapidity with which the bodies can generate force. The stiffer the body, the quicker its response to an impact and the higher the impact force. The anvil has a much higher spring rate than the steel strap in Fig. 3–28D.

Impulse forces are also affected by energy losses during collision. It may take a high force to deform a body during the compression phase of an impact, but the body may not return this force during the extension phase (thereby reducing the total impulse).

Forces generated in an impact are usually extremely high. Effects can be seen in high-speed photographs of baseballs or golf balls being distorted by impact with a bat or club. If such distortions



Drawings courtesy of MACHINE DESIGN Magazine: May 9, 1968: pp. 165–172

Fig. 3-28. Impact from a hammer can be stopped rapidly by an anvil (view A), or slowly by a springy metal strip (view D). Areas under both force-time curves are equal for equal hammer blows, but maximum stopping forces and stopping times differ, as shown in views B and C.

were produced with static forces, tons of force would be required. Yet these forces can be produced with a bat or club weighing only a few pounds. Because impact forces are so large, the forces produced on the colliding bodies by such factors as body weight, drive springs, or magnetic fields during the impact can usually be ignored.

Translation Plus Rotation

Collisions between translating bodies aimed at the centers of gravity are hard enough to predict numerically, but the situation is made more complex if one or more of the bodies is rotating before or after the collision.

If two rotating bodies collide so as to produce only a change in rotation in each, their motion can be described by equations which are the exact analogues of the basic linear equations. Inertia replaces mass, angular velocity replaces linear velocity, and torque replaces force. The impulse, momentum, and energy equations are:

$$\int_{t_0}^{t} \tau \, dt = I_1\omega_1 - I_1\omega_1' \qquad (3\text{–}9)$$

$$I_1\omega_1 - I_1\omega_1' = I_2\omega_2' - I_2\omega_2 \qquad (3\text{–}10)$$

$$\tfrac{1}{2}I_1\omega_1^2 + \tfrac{1}{2}I_2\omega_2^2 = \tfrac{1}{2}I_1\omega_1'^2 + \tfrac{1}{2}I_2\omega_2'^2 + Q \qquad (3\text{–}11)$$

and energy loss is:

$$Q = \tfrac{1}{2}(1 - K^2)\,\frac{I_1 I_2}{I_1 + I_2}\,(\omega_1 - \omega_2)^2 \qquad (3\text{–}12)$$

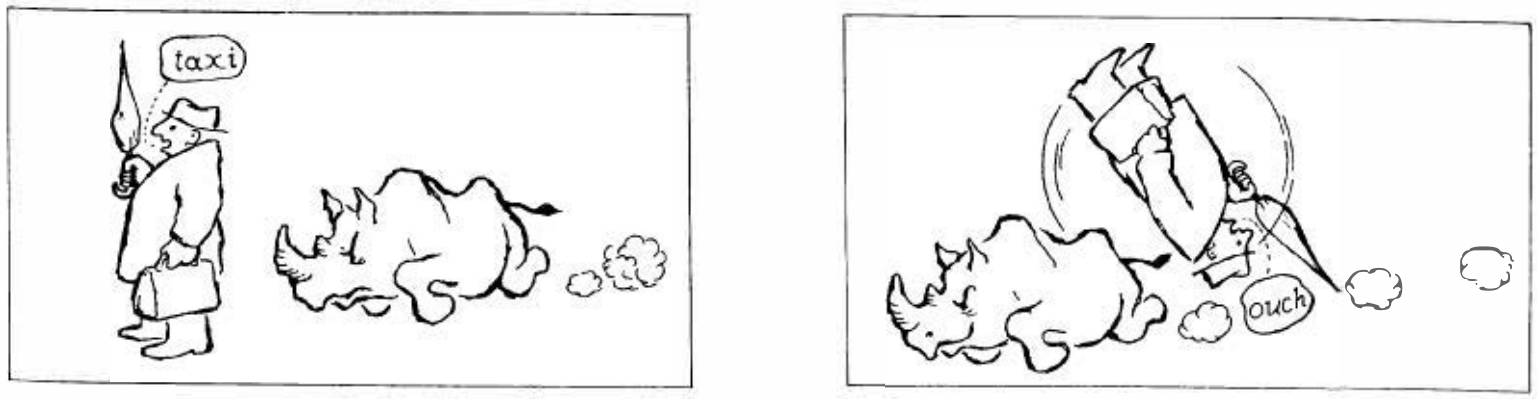


Fig. 3–29. Linear momentum is not necessarily conserved; neither is angular momentum. Linear momentum can be converted into angular momentum as shown here, or vice versa, to conserve total momentum in the system.

Unfortunately, rotary impacts rarely involve only a change in rotation. In fact, even translational collisions are seldom aimed so perfectly at the centers of gravity that they cause changes only in linear motion. In most actual cases, a mixture of linear and rotational motion is involved. For example, if a translating body strikes another at some point other than the center of gravity, part of the initial kinetic energy is converted to rotational kinetic energy, Fig. 3–29; the rest goes to translational kinetic energy or is lost through internal friction, deformation, external friction, vibration, or heat, as with the translating bodies considered earlier.

The basic momentum, energy, and impulse equations become more complex to account for this additional degree of freedom of the colliding bodies:

$$\int_{t_0}^{t} \tau_1 \, dt = I_1\omega_1 - I_1\omega_1' \qquad (3\text{–}13)$$

$$\int_{t_0}^{t} \tau \, dt = I_2\omega_2 - I_2\omega_2' \qquad (3\text{–}14)$$

$$\int_{t_0}^{t} F \, dt = m_1v_1 - m_1v_1' = m_2v_2' - m_2v_2 \qquad (3\text{–}15)$$

$$\tfrac{1}{2}m_1v_1^2 + \tfrac{1}{2}m_2v_2^2 = \tfrac{1}{2}m_1v_1'^2 + \tfrac{1}{2}m_2v_2'^2 + \tfrac{1}{2}I_1\omega_1'^2 + \tfrac{1}{2}I_2\omega_2'^2 + Q \qquad (3\text{–}16)$$

If a colliding body is mounted on bearings, it is unable to translate when struck. Additional energy is lost as the body responds to the blow. By the same token, a body confined by slides, Fig. 3–30, loses energy if struck along a line that does not pass through the center of mass, because it cannot rotate. To reduce wear and heat in the constraints, all impacts should be along a line through the center of gravity and parallel to the slide surface.

Rotary bearing reactions can be almost eliminated if the rotating body is struck at a point called the "center of percussion," Fig. 3–31. Some bearing reactions, of a small order of magnitude, occur even when rotary bodies collide at their centers of percussion, but the large reactions are eliminated. There is a different striking point, or center of percussion, for every bearing location in the body, with one exception: there is no center of percussion corresponding to a bearing through the center of gravity of the body. Therefore, striking a wheel always produces bearing reactions.

The center of percussion exists only for rigid bodies where the coefficient of restitution is the main influence on impact energy loss. If the body can acquire significant vibrational energy, Fig. 3–32, it will not behave as a body struck at the center of percussion. There will be bearing reactions and frame and vibration losses.

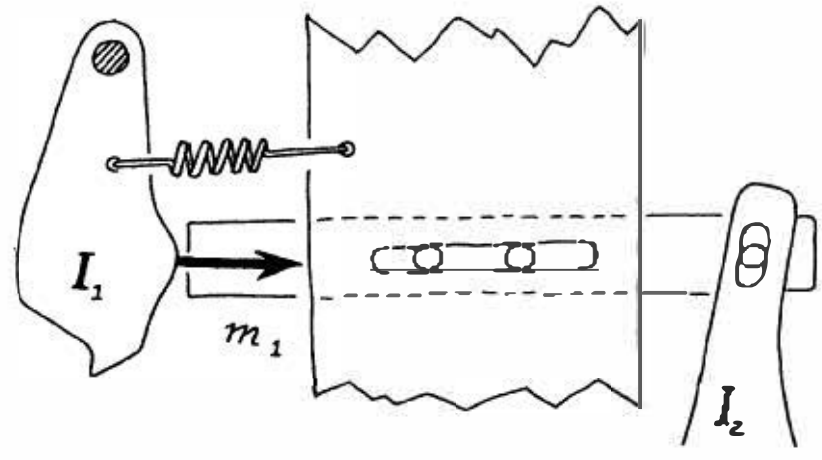


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Fig. 3–30. A great deal of energy will be lost in bearings and slide mounts if impacts are not directed through the centers of gravity of colliding bodies. m_1 is the mass of the sliding member; I_1 and I_2 the moments of inertia of the rotating members.

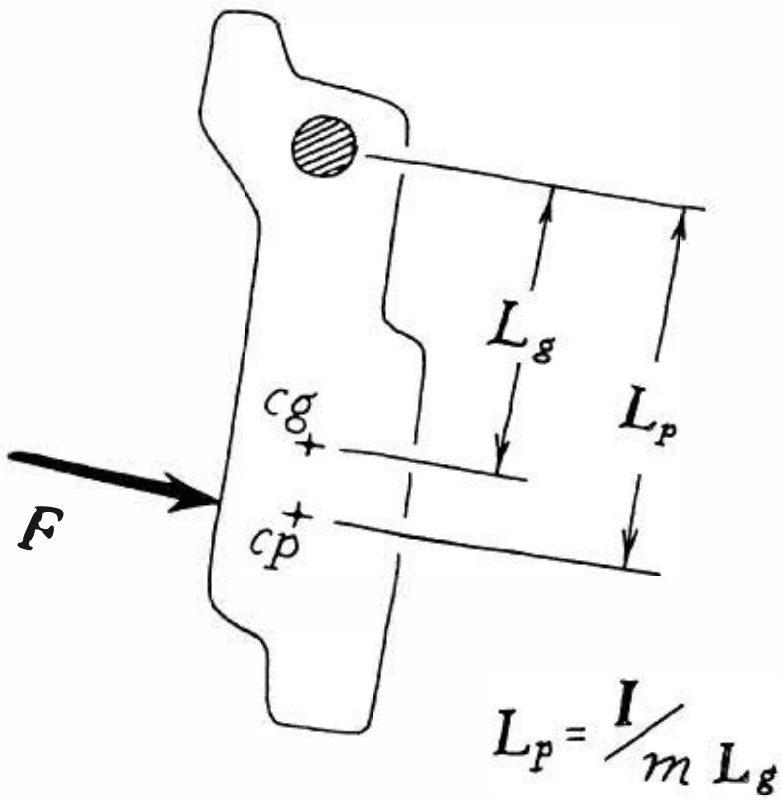

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Fig. 3–31. Relationship between center of gravity (cg), and center of percussion (cp). F is a force; L_g and L_p distances from the center of rotation; and m and I the mass and moment of inertia of the link.

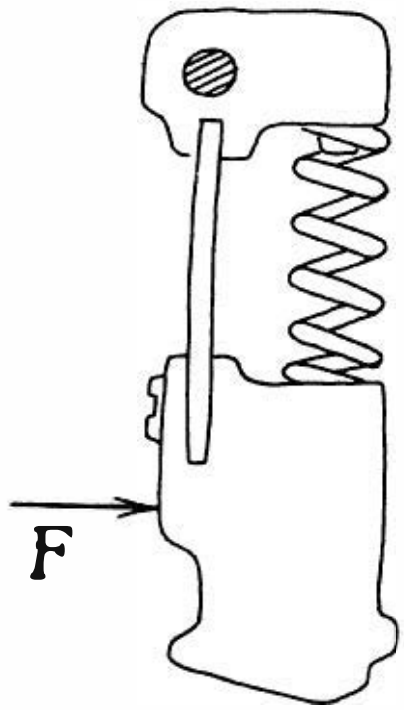

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Fig. 3–32. A non-rigid body, subject to internal energy losses, has no center of percussion. Any applied force, such as F, will cause bearing reactions.

Conservation of Momentum

Most designers have been carefully schooled in the theories of conservation of momentum. We know that linear momentum is conserved when two flying bodies strike each other. We expect, therefore, to find that angular momentum is conserved when two rotating machine bodies collide. The sum of the angular momentums in the two bodies before collision, should then be the same as the sum of angular momentums after collision. However, this is not the

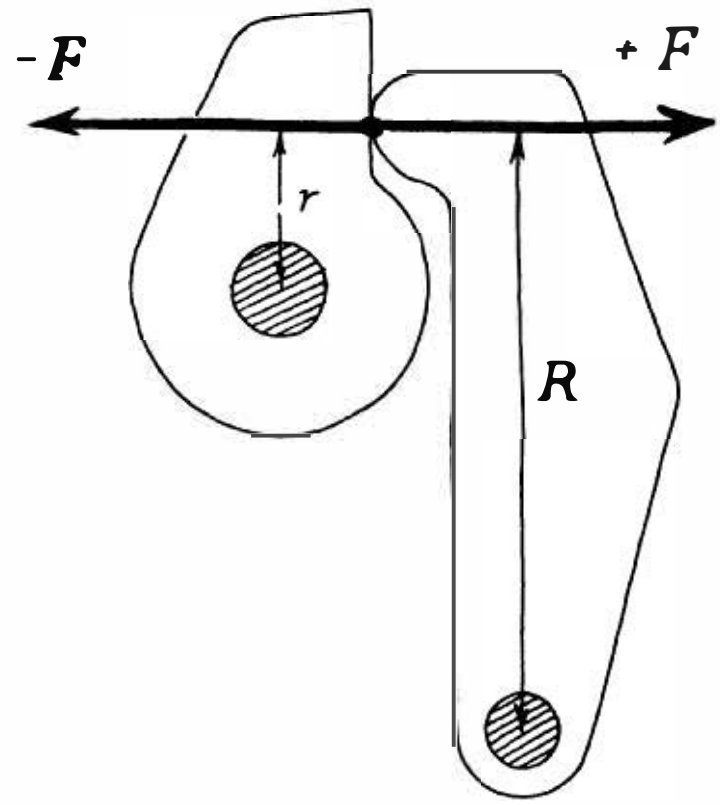

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Fig. 3–33. Angular momentum is not conserved in a collision between rotating bodies striking at unequal radii (r and R). The forces ($+F$ and $-F$), produced by such a collision are equal and opposite; the torques are not.

case. To see why, consider the rotational analogue of the linear impulse momentum equation. This might be called the Law of the Conservation of Angular Momentum.

Angular impulse equals change in angular momentum:

$$\int_{t_0}^{t} \tau\, dt = \int_{t_0}^{t} FR\, dt = I_1\omega_1 - I_1\omega_1' \quad (3\text{–}17)$$

This law says that an external impulse is required to change the momentum in a body, and this is always true. In the linear case, it was shown that the impulses exerted by two colliding bodies on each other were equal, and that momentum was, there-

fore, conserved during internal impacts in a linear system. Rotating bodies in collision exert equal and opposite forces on each other for equal times, just as translating bodies do. (See Fig. 3–33.) But unless the rotating bodies have equal radii, these forces produce unequal torques. Different torques, acting for the same time, produce different angular impulses and, therefore, different changes in momentum in the two bodies. What one gains, the other does not necessarily lose, and angular momentum is not conserved.

Actually, of course, each body experiences bearing reactions during the impact (or shortly thereafter if the bodies are struck at their centers of percussion). As a result of impact, therefore, the frame (Earth) exerts torque on the two-body "system." Only by taking this torque into account could the total change in angular momentum of either body be rigorously explained. Since bearing reactions are nearly impossible to calculate, this can seldom be done. And, in fact, there is really no such thing as a Law of Conservation of Angular Momentum; there is only the Law of Conservation of (Total) Momentum, Angular and Linear.

As a result of all this, a useful analogue of the basic two-body momentum expression (Equation 3–1) can be written only for a few special cases. For example, if the bodies have radii R_1 and R_2 and the collision is purely tangential, the momentum equation can be written:

$$\frac{I_1\omega_1 - I_1\omega_1'}{R_1} = \frac{I_2\omega_2' - I_2\omega_2}{R_2} \qquad (3\text{-}18)$$

The coefficient of restitution used in the corresponding energy equation should probably be based on the change in relative tangential velocities of the two bodies. Actually, the original coefficient becomes less meaningful as the impact becomes more complex; it is more important to worry about energy loss than about K.

When an impact involves both radial and tangential velocities, the energy and momentum equations are further complicated because more than one coefficient of restitution is involved. In general, the linear and rotational impulse equations (3–13 through 3–16) must be solved for each body.

In summary, collisions between two rotating or rotating and translating bodies involve the same uncertainties of force level and energy loss as do pure translational collisions, but they are further complicated by the final distribution of energy between rotational and translational modes and because angular momentum is apparently not conserved during impacts between two rotating machine members. The presence of bearings or other constraints must also be taken into account.

Multiple Impact

Impact between two machine members almost always consists of more than one collision. This seriously complicates the prediction of motion and energy losses. Figure 3–34 shows a large spring driven link striking a smaller one. Obeying the equations, the striker pauses, or rebounds, while the small link flies away from the point of impact. The spring driving the large link soon stops the reverse motion, and again starts the large link moving toward the small link. (Rebound kinetic energy has been converted to potential energy in the drive spring, then reconverted to kinetic energy in the forward direction.) The driver again catches the

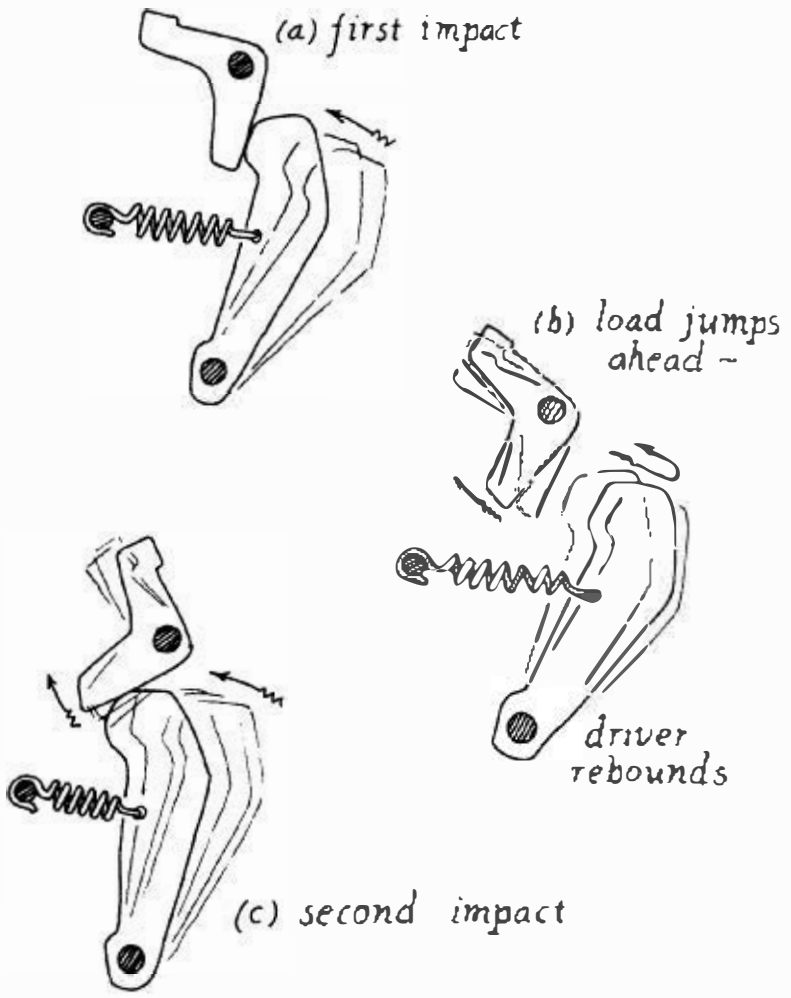

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Fig. 3-34. Impact between machine elements usually consists of more than one collision. A different amount of energy is lost each time.

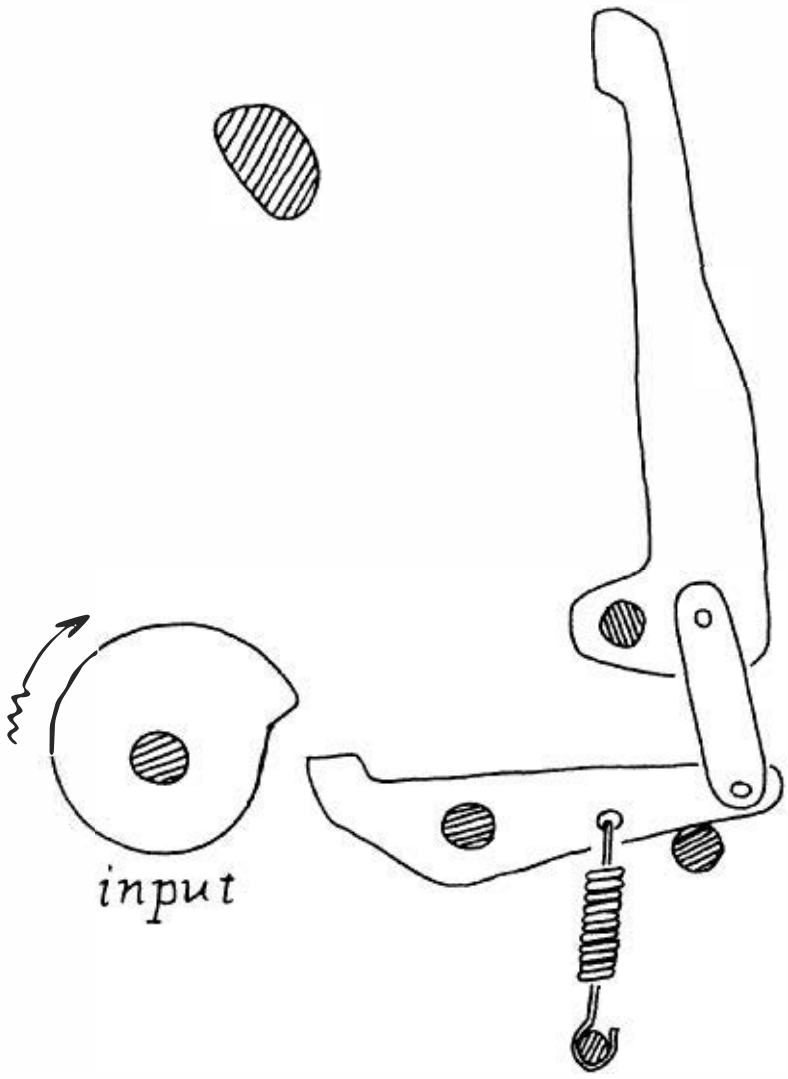

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Fig. 3–35. Collisions involving several bodies are nearly impossible to calculate.

coasting small link and restrikes it. The rebound-stop-return-restrike process is repeated.

A typical impact involves three or more collisions before the two bodies finally move together as one. Some energy is lost in friction (internal and external), vibration, distortion, and bearing reactions with each blow. Since each blow takes place at a different impact energy level (or impact velocity) and with different contact geometrics (more friction loss occurs in the second contact than in the first, for example), a different percentage of energy is lost each time. To calculate final velocities and impact forces and times, energy losses or impulse forces would have to be predicted for each collision. This is essentially impossible, even when only two bodies are involved.

Multiple collisions per impact is not the worst problem. More than two bodies are usually involved in most machine collisions, as seen in Fig. 3–35. Each is struck, moves forward, strikes the next, rebounds to be restruck, moves ahead again to reinforce the next or to be itself struck by the next which is rebounding, after having struck the third, etc. Many collisions occur between link pairs until the whole train moves forward together. The action cannot be predicted accurately.

Maximizing or Minimizing Impact

Sometimes, impact effects must be minimized to reduce contact forces, extend bearing life, or reduce bounce. Other applications require maximizing of impact effects to develop a high force with a small body (a typewriter key, for example). Or it may be necessary to use impact to transfer energy quickly from one body to another. Although the impact equations cannot really be solved, they can be used; along with a general knowledge of impulse, energy losses during impact, and location of center of percussion to determine desirable design directions.

For example, suppose it is necessary to maximize the kinetic energy delivered to a struck body (m_2), in a linear impact. Equations 3–1 through 3–5 verify the design directions that are somewhat intuitive: maximize the impact velocity and the mass of the striker (m_1).

Fig. 3–36. The golfer's paradox.

Table 3–1. How to Control Impact

In order to. . .	Maximize	Minimize	Comments
Maximize velocity of struck body (m_2 or I_2) after impact	K, $\frac{m_1}{m_2}$, $\frac{I_1}{I_2}$, m_1v_1, $I_1\omega_1$	Q, $\frac{m_2}{m_1+m_2}$, $\frac{I_2}{I_1+I_2}$	m_2 should have minimum negative velocity when struck. Strike m_2 at cg. Strike I_2 at cp.
Minimize velocity of struck body (m_2 or I_2) after impact	Reverse of above		
Minimize time required for two bodies to start moving as one	Q, $\frac{m_2}{m_1+m_2}$, $\frac{I_2}{I_1+I_2}$, m_1, I_1	K, (v_1-v_2), $(\omega_1-\omega_2)$, m_1v_1, $I_1\omega_1$	Might be important in a switching relay.
Maximize impact force (F)	K, spring rate of both bodies, m_1v_1, m_2v_2, $I_1\omega_1$, $I_2\omega_2$	Q	Useful in printer design.
Minimize impact force (F)	Reverse of above		Reduces stress, strain, wear.
Minimize rotary-bearing reaction forces	Q	K, body spring rates, frame stiffness, $I\omega$'s, clearances	Strike all bodies at cp. Reduces stress, strain, wear.
Minimize slide-bearing reaction forces	Q	K, body spring rates, frame stiffness, Mv's, clearances	Strike sliding bodies at cg. Reduces stress, strain, wear.
Maximize energy loss during impact (Q)	$\frac{m_2}{m_1+m_2}$, $\frac{I_2}{I_1+I_2}$, mv's, $I\omega$'s, (v_1-v_2), $(\omega_1-\omega_2)$, I_1, m_1	K, stiffness of bodies and constraints	Used to convert kinetic energy to deformation energy in a forging operation.
Minimize energy loss during impact (Q)	Reverse of above		Strike sliding bodies at cg, rotating bodies at cp.
Maximize rebound velocity of striker (m_1 or I_1)	(v_1-v_2), $(\omega_1-\omega_2)$, $\frac{m_2}{m_1+m_2}$, $\frac{I_2}{I_1+I_2}$, K, v_1, ω_1	Q, m_1, I_1	m_2 should be at rest or moving toward m_1.
Minimize rebound velocity of striker (m_1 or I_1)	Q, m_1, I_1, v_1, ω_1	(v_1-v_2), $(\omega_1-\omega_2)$, $\frac{m_2}{m_1+m_2}$, $\frac{I_2}{I_1+I_2}$, K	m_2 should be at rest or moving away from m_1.

Another requirement may be to minimize the energy lost during the collision. If m_1 is the mass of the striker, and m_2 is the mass of the struck body, then this can be accomplished by maximizing the coefficient of restitution and by minimizing the mass ratio $m_2/(m_1+m_2)$. Final velocity v_2' produced in m_2 can be increased by maximizing the ratio m_1/m_2.

To accomplish the same results in a rotary impact, first decide which set of rotary equations to use. Usually, the simplest set (3–9 through 3–12) will give sufficient insight if their limitations are kept in mind. To maximize final kinetic energy in a rotary impact, it would be necessary to maximize the inertia and angular velocity of the striker, maximize K, maximize the ratio I_1/I_2, and minimize $I_2/(I_1+I_2)$. Also, both bodies should be made reasonably rigid

to reduce vibrational losses, and collisions should be made to occur at the centers of percussion of both bodies to minimize frame losses.

In more complex rotary collisions, more complex equations are usually needed to refine the first assumptions. Equations 3–13 through 3–16 often can be used to handle such situations.

Table 3–1, may help the designer determine which way to go to produce desired results.

The Paradox of the Golfer

Impact can produce "impossible" forces, as anyone who has ever seen high-speed pictures of the impact between golf club and golf ball can attest. Calculations and measurements indicate that the contact time between ball and club is approximately one-half of a millisecond and that the average force is 1500 lbs during this interval. This means that the maximum force is closer to 2,000 or 2,500 lbs; and all of this generated by a man who weighs perhaps 200 lbs and a club weighing a few ounces, Fig. 3–36.

In addition, Newton has assured us that whenever we apply a force to a thing, the thing applies an equal and opposite force to us. If the man and his club produce a force of 2,000 lbs on the ball, then that little 1⅞-inch diameter plastic and rubber sphere simultaneously must be applying a force of 2,000 lbs on club and man! And indeed it is. How can this be? Surely a force of 2,000 lbs is enough to throw the man all the way back to the clubhouse. Why does only the ball move? The answer, of course, lies in the fact that the impulse (force-times-time) applied to the man is too small to move him; at least for any distance. He is a large mass with considerable internal damping and is virtually unaffected by this "collision" with a small elastic body. By the same token, many machine members are not affected by repeated collisions that generate, at first glance, what appear to be destructive forces. Learn how to predict the forces, but do not be unduly concerned about them. Understand the impact process and learn to use it and live with it.

CHAPTER 4

The Mechanics of Intermittent Motion

A SPRING-DRIVEN RATCHET

The Problems of Intermittent Motion Mechanisms

Now that we are experts on graphical and elastic body mechanics, let us take a closer look at a special topic—intermittent motion devices.

It would be nice if all intermittent motion mechanisms were as easy to analyze as a "block-on-the-table" or the "finger ratchet," studied earlier. Unfortunately, however, this is rarely the case. In most machines, the driving and driven linkages are considerably stiffer than fingers. Drive velocities are high, and contact times between parts can be very short and extremely difficult to measure. The result: impact, vibration, chatter, surprisingly high contact forces, large energy losses, instability in the machine, and other horrors. Let us take a look at an actual situation.

Spring-driven Ratchet Mechanism

Figure 4–1 shows progressive stages of a ratchet mechanism in operation and corresponding plots of the net torque on the ratchet wheel as a function of time during a single drive cycle. In the first stage (A), only the drive cam is in motion and the net torque on the wheel is zero. In the second stage (B), the cam has released the arm and the drive spring has moved the arm forward to contact the wheel. All the energy which is stored in the arm during this pre-travel is now dumped rapidly into the wheel and into the wheel bearing. The arm produces a very high force on the wheel for a short period of time. This creates drive torque and a bearing reaction force F_B, which, in turn, produces a friction force F_b. This blow inevitably causes the wheel to jump ahead of the arm so that, for a brief instant, the force exerted by the arm on the wheel disappears.

By the third stage (C), however, the arm (still driven by its spring) has caught up with the wheel (because in the meantime the wheel has been slowed by its load and also by bearing friction) and the arm has delivered another, lesser blow to the wheel. The wheel has jumped ahead again, the arm has caught it again, and struck it a third and a fourth time. Following the fourth collision, the arm and the wheel finally travel along together so that the arm exerts a fairly constant force on the wheel (we are assuming a drive spring with a fairly flat rate). The force levels and times associated with each blow are different than those associated with each previous blow, in general, becoming lesser as the two parts get together. There is some opinion that a "typical" machine situation involves about five separate impacts, each time that two links collide. Theoretically, there are an infinite number of separate impacts in each collision, but our old friend "energy loss" does make it possible for the two parts eventually to get together.

In the fourth stage (D), a so-called non-overthrow tooth on the ratchet arm has engaged the inner row of teeth on the ratchet wheel, exerting a sudden, and very sharp, stopping force on the wheel (and producing bearing reactions as before). The net torque

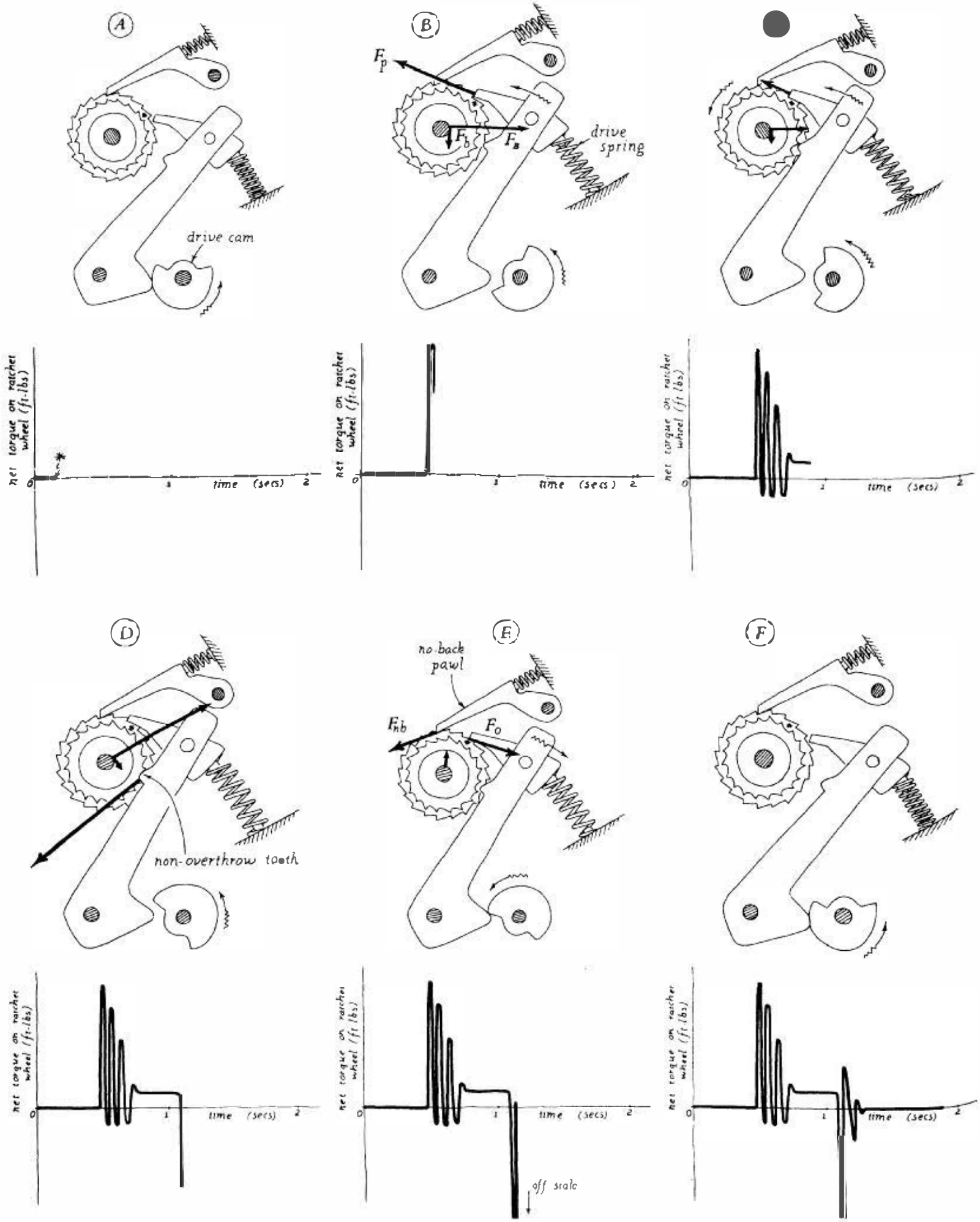


Fig. 4–1. Progressive stages of a ratchet mechanism in operation showing step-by-step development of the torque-versus-time curve for this impacting device.

is negative now and the wheel comes to a stop. The no-back pawl prevents significant rebound but this stop, just as the drive impact, would consist of several short, sharp blows rather than one long one.

As in the simple case of the finger ratchet studied earlier, the positive area under the torque–time curve must be cancelled by an equal negative area if the wheel is to be brought to a complete stop. A little reflection will show that the time required to stop, in this situation, must be considerably shorter than the drive time. In an actual machine the drive time might be a second or more, as shown on the graph, and the stopping time, a small fraction of a second. For the curves to generate equal positive and negative areas, therefore, the stopping torque must be very large, as shown in curve (E).

In stage (E), the ratchet arm has started to retract, driven by the drive cam. The arm, in this condition, exerts a negative drag on the ratchet wheel, but this drag is counteracted by an equal and opposite reaction from the holding pawl, thus the net torque on the wheel is zero and the wheel remains at rest. In stage (F), the net torque on the wheel is still zero. The arm is ready to pick up and drive the next tooth of the wheel.

Calculations—Spring-driven Ratchet

For a real understanding of the ratchet mechanisms, acceleration; velocity; and displacement curves for the wheel as a function of time should be drawn. To draw accurate curves, however, values must be put on the torque curve just sketched in. To do this, the maximum torques, impact and motion times, etc., must be known.

Obviously, these values will be extremely difficult to obtain. Start, perhaps, by drawing a curve of the torque on the drive arm as a function of time. We now have a pretty good understanding of the influence of the drive spring on the arm; since we design the spring (and therefore know just how much force it can produce), and since the spring operates for a fairly long and easily measured time. We know, therefore, what the momentum of the drive arm is when it contacts the ratchet wheel, and can use this to estimate the approximate magnitude of the impact forces when it encounters a stop; similar to the situation illustrated in Figs. 3–1, 3–4, and 3–5.

But the wheel is not a stop. It is free to move and, in fact, is intended to move under the influence of the blow exerted on it by the drive arm. As a result, the arm will lose some, but not all, of its energy when it strikes the wheel--some of this energy going into useful work to drive the wheel; some being lost in bearing and internal friction. The arm will then gain more energy from its drive spring before striking the wheel a second time; and will then let loose a second bundle of energy. It is doubtful that there is a man alive who could calculate these energy losses and transfers.

If the time involved in the multiple impacts was known, on the other hand, impact forces could presumably be computed since we know that the area under the stopping portion of the torque curve must cancel the area under the drive portion of the curve. But again, we hit a snag. We could estimate the time involved in a total collision process, but to measure peak forces, the time for each and every one of the four or five little separate impacts that actually occur during one collision must be known.

Of course, if we could experimentally measure or estimate the maximum forces involved impact times could then be estimated, but this is very difficult also. Impact specialists tell us that the forces generated at the surfaces of colliding bodies are not transmitted equally to all other parts of the body; because we are dealing with elastic bodies, not rigid bodies. What happens is that stress waves move through the body, reflect from bearings or free surfaces, return toward the source of impact, magnifying or cancelling subsequent stress waves (from other individual impacts) that have been generated in the meantime and are themselves surging through the body. This situation makes it virtually impossible to measure impact force levels, even if strain gages were mounted on the colliding bodies.

In fact, there appears to be little that can be done to put actual values on the graph of torque as a function of time which has been drawn for this simple ratchet situation. Mathematically inclined designers are in the same boat, however. They can write equations for this situation and can produce numerical answers, but only by *assuming* impact times or maximum force durations, or spring rates, etc., for the colliding bodies. We can do this too. If you are designing the ratchet, you will know what its inertia is and how stiff the drive and pawl springs are. If you assume that each impact involves a force–time relationship that is sinusoidal (remember the quasi-static model), and that total time duration is on the order of magnitude of a few milliseconds (a good assumption in many machines), also that

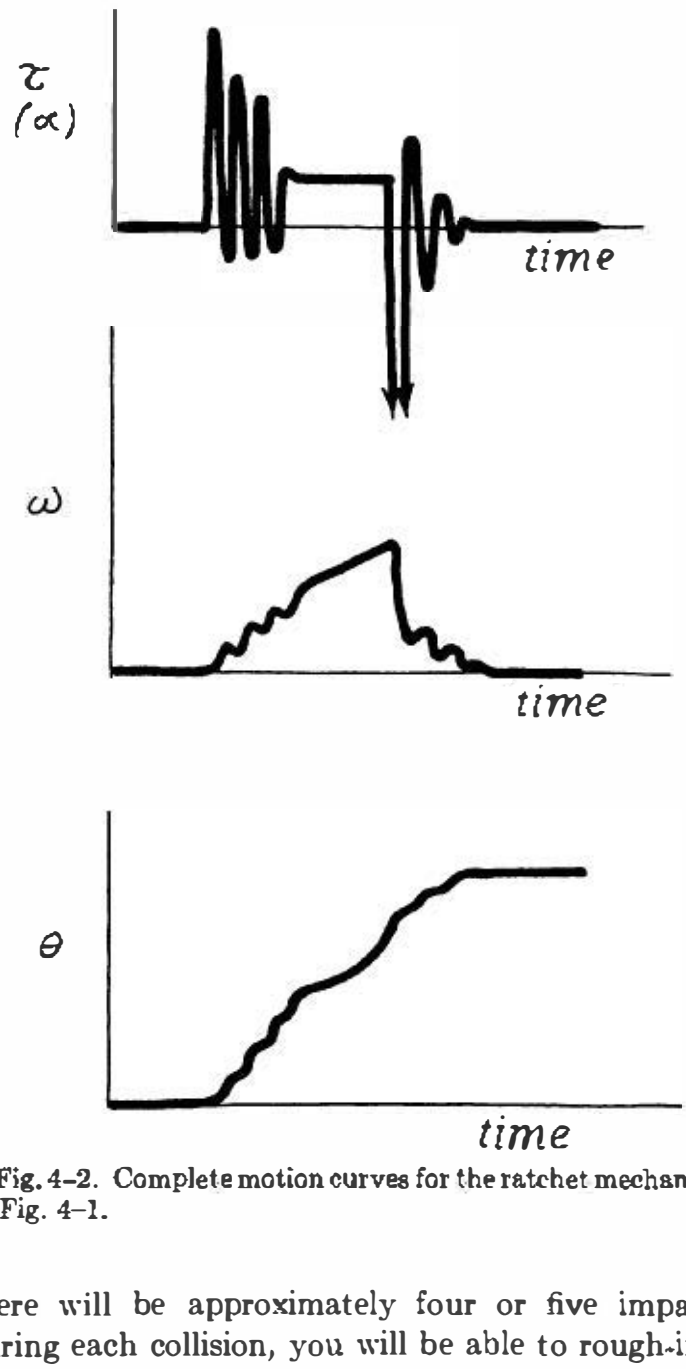


Fig. 4–2. Complete motion curves for the ratchet mechanism of Fig. 4–1.

there will be approximately four or five impacts during each collision, you will be able to rough-in a torque–time curve that is as good an approximation to the actual as the world's finest "calculator" could achieve.

Using graphical techniques we will now approximate the acceleration, velocity, and displacement curves for the ratchet wheel as well. The resulting curves are shown in Fig. 4–2.

THE PROBLEM OF FORCE MAGNIFICATION

We have run into a serious problem in trying to calculate the exact force, acceleration, velocity, and displacements of an intermittent motion device when collisions are involved. The analysis, however, has revealed other problem areas that are perhaps more important to us as machine designers than the purely theoretical problem of being able to put exact numbers on the various functions involved. For example, we have again exposed the problem of force magnification first considered in Chapter 3. Even though we cannot calculate exactly what the peak impact forces are, we can see that they can be many, many times greater than the drive forces which we have thoughtfully provided for our machines—structures and linkages intended to survive only under drive-force levels would fail very rapidly in many intermittent motion mechanisms. Increased stiffness and strength is most easily accomplished by an increase in the mass of the parts and structure involved. But we have seen that increases in mass tend to increase force levels during impact, defeating the purpose of the mass increase. It is necessary, therefore, to do everything possible to provide increased strength with a minimum increase in mass. Machine design experience shows that this can be done in most situations so that the various parts are capable of withstanding the forces involved.

But the problem goes beyond that of designing parts that are strong enough not to break down if subjected to operating forces. Usually the concern is about the cost of the machines; and infinite strength and stiffness cost money, especially if mass must be minimized. Furthermore, even if machine members are strong enough not to fail catastrophically, magnified forces can lead to serious problems of wear and instability, as will be seen in the next chapter. Higher forces should not just be compensated for then, we should try to avoid them. And the only way to do this is to avoid sudden changes in velocity.

The ratchet just studied has built-in sudden-changes-in-velocity, of course. The wheel was at rest when struck by a rapidly moving drive arm. Now the wheel must suddenly start to move and the arm suddenly slow down when they collide. This is inherently a questionable design; this type of mechanism should only be used where loads are very light and where low cost is very important. Under these conditions, of course, spring or coil-driven ratchets can give excellent service; witness the highly reliable stepping switch (Fig. P–2 in the Preface, for example). But whenever possible, avoid impact! Or at the very least, impact velocities should be kept as low as possible.

Reducing Forces by Reducing Impact Velocity

One way to reduce impact velocity is to design the ratchet in such a fashion that the driver acquires

a high velocity as soon as possible during the pre-travel, but has a relatively low final velocity, as shown in Fig. 4–3. This can be done by proper design of a cam, perhaps, or a drive solenoid or spring system. We are usually interested in a high *average* velocity in an intermittent motion drive system so that a large number of operating cycles can be produced in a given time. A high average can be achieved by using a high initial acceleration and a low final or impact velocity will still remain, as shown in 4–3A. On reflection, this is not a common approach, but it is an excellent way to improve the performance of a high-speed intermittent motion machine.

Impacts generated in a driver are also transmitted through an entire machine. They will be magnified further wherever there is clearance, backlash, pre-travel, etc., involved. Keep the whole machine as "tight" as possible; nothing is more effective in reducing wear in an intermittent motion situation.

Another method of reducing impact is to use a system in which the driving member approaches the driven member at an angle, when first making contact. This results in a low RELATIVE velocity between colliding surfaces at the time of impact, even if the driver is moving at a high rate of speed. And it is relative velocity, not absolute velocity that determines the magnitude of impact forces, as was seen in the previous chapter.

The Geneva mechanism, shown in Fig. 4–4, works

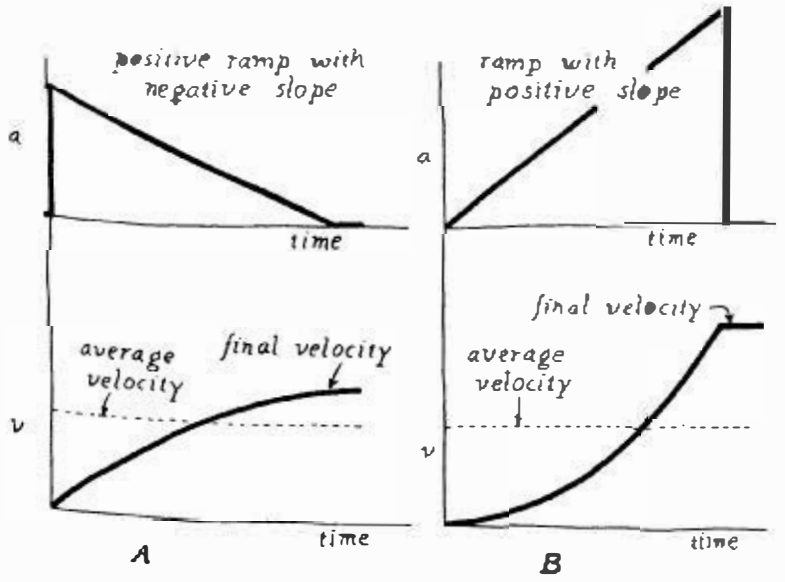


Fig. 4–3. If the acceleration curve is a positive ramp with a negative slope it will produce a lower final velocity than would an acceleration curve with a positive slope, even though both produce the same average velocity and, therefore, the same cycle time. A negative slope torque (acceleration) applied to a ratchet arm, for example, will produce a lower impact velocity without increasing operating time.

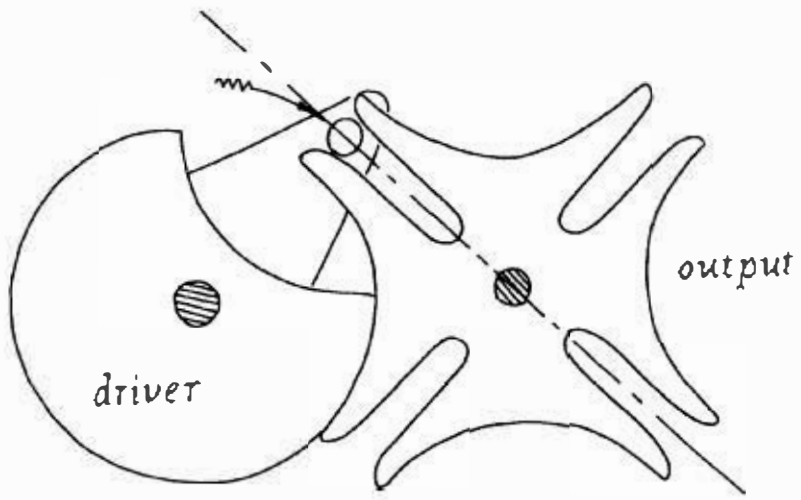


Fig. 4–4. The drive pin in a Geneva mechanism enters the drive slot parallel to the wall of the slot to reduce impact velocity.

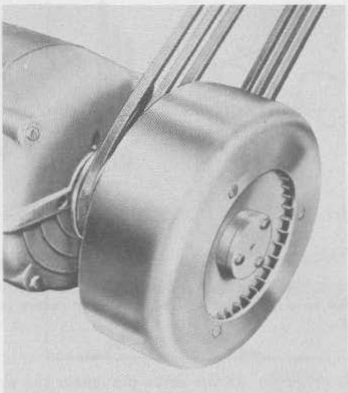
on this principle. As the drive pin enters the slot in the stationary output member the pin is moving along, tangent to the centerline of the slot, with zero relative velocity between the pin and the wall of the slot. As the driver continues to rotate, the pin moves against the wall of the slot and picks up and indexes the output member. The change in velocity in the output is fairly gradual. True impact is not involved if the parts are made correctly. Genevas will be studied more thoroughly in a later chapter.

This type of mechanism, incidentally, where the driver approaches the load at an angle, produces less output motion for a given input stroke than does a direct push device, such as a ratchet. But again, this is an excellent way to reduce impact stress levels.

Programming Drive Forces to Reduce Stress

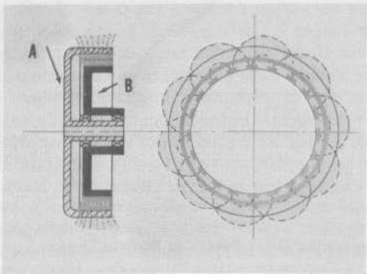
Another method of reducing stresses caused by collisions is to design the power plant so that it pushes the driver link gently until contact has been made with the output link; and then pushes hard. This will probably increase the time required for the driver to move from its cocked position to drive position (the pre-travel time) but could have significant benefits in the life of the machine. Again, this is not a very common approach, but it certainly has a lot to recommend it if this complexity can be afforded. There is one rotary solenoid manufacturer who has manufactured solenoids that produce a "soft" followed by a "hard" force of this type. Cam systems can also be used effectively to change the leverage of a driver, for example, during the stroke.

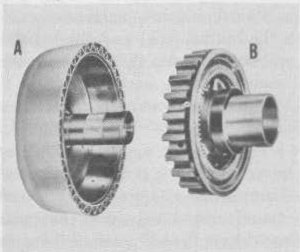
Pneumatic and hydraulic drivers can be designed



Photograph courtesy of Fairchild Hiller, Industrial Products Division

Fig. 4–5. Eddy current coupling used to program the torque delivered to a load and thus reduce the rate-of-change of acceleration when starting and stopping. Also see Fig. 4–6.





Photograph courtesy of Fairchild Hiller, Industrial Products Div.

Fig. 4–6. Further details of the eddy current coupling used to reduce accelerations in a drive system. Also see Fig. 4–5.

to produce tailored forces, also. In systems involving clutches, brakes, electric motors, and the like (Chapters 13 and 15, for example), we can reduce peak torques applied to a load by using devices such as the eddy current coupling shown in Figs. 4–5 and 4–6. This coupling will slip until sufficient velocity is attained by the driver to generate a drive torque in excess of the load torque. By programming the driver speed, the rate at which torque is applied to the load can be programmed.

Programming the Velocity of the Driver

A system can also be designed in which the velocity of the driver is programmed to reduce impact

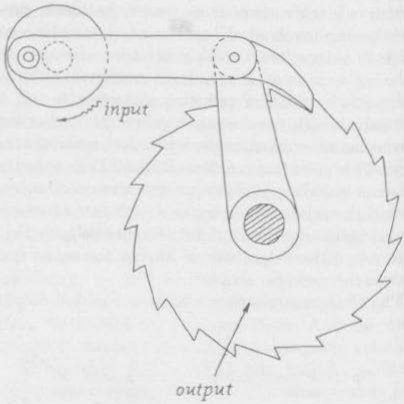
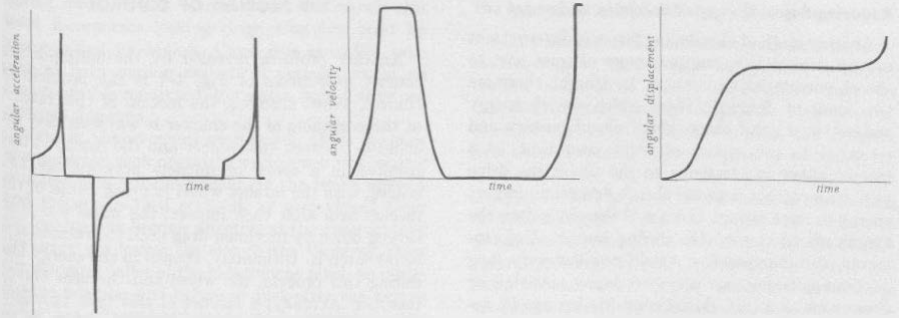


Fig. 4–7. Ratchet mechanism driven by a four-bar linkage. Also see Fig. 4–8.

velocity. An example is shown in Fig. 4–7, where a ratchet is driven by a cam rather than by a spring, as in the first ratchet example. The resulting acceleration, velocity, and displacement of the ratchet wheel is shown in Fig. 4–8; this is quite an improvement over that in Fig. 4–2.

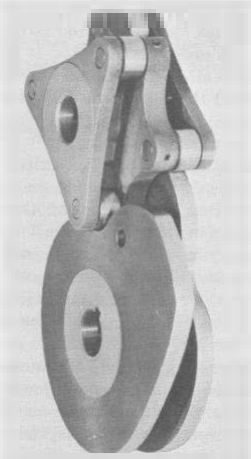
Cam Systems—Programmed Displacement

This idea of programming velocity is applied to many intermittent motion mechanisms, even to those that do not normally involve impacts. Cam systems, for example, are carefully designed to avoid sudden changes in velocity of the driven member, to reduce



...wn in Fig. 4-7. A ... is type o... smoothly ... hat ... Fig. 4-11 becaus... impacts ... vebeen reduced or eliminated by the "programmed" drive system.

inertial loads, and hence, to reduce force magnification. Generations of cam designers have struggled to find the "perfect" cam profile for an indexing mechanism, and still argue heatedly in support of their favorite... the driver usually ... in a consta... y but ... shaped in such a way as to program the displacement of the follower with which it is always in contact. In a well-designed system the follower starts at zero, is accelerated gradually, and is decelerated gradually. Ideally, initial and final acceleration curves have zero slopes to eliminate jerk. A typical cam indexer is shown in Fig. 4-9. Three popular motion curves for cam systems are shown in Fig. 4-10.



Photograph courtesy of ... Cam and Machine Company

...g. ... impacts ... cam system such as this. See Chapter 8 for further details.

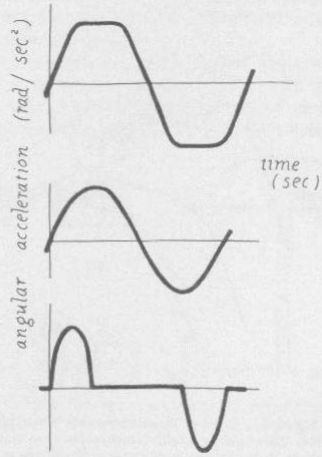


Fig. 4-10. Examples of the type of motion curves which ... be obtained with well-designed cam systems.

Reducing Forces through Absorption of Energy

Another method of reducing forces in intermittent motion devices is to use a damper of some sort to absorb energy during impacts. In general, there are two kinds of dampers: those which absorb energy and retain it; and those which absorb energy and return it to the system at some later time. If a marshmallow was fastened to the tip of the drive pawl on a ratchet it would absorb and retain impact energy to some extent, and would thereby reduce the magnitude of stress, etc., during impact. Unfortunately, the marshmallow would not last very long in this application nor are there many materials or devices which would. Sometimes, friction can be deliberately built into a mechanism train to absorb energy in this manner, but friction is very unpredictable. Besides, any sort of frictional energy absorption can lead to wear. Energy absorbers of the "permanent" deformation type are used for structure isolation only and are not used in power transmission systems.

It is possible to provide a damper which absorbs and returns energy as would the spring shown in Fig. 4–11, but the designer should realize that such energy *will* be returned to the system, and that if it is returned in phase with a new load or drive-peak torque, the "damper" would magnify, not reduce, stress levels. See Fig. 3–22 which shows what can happen in a mechanism train when elasticity is uncontrolled.

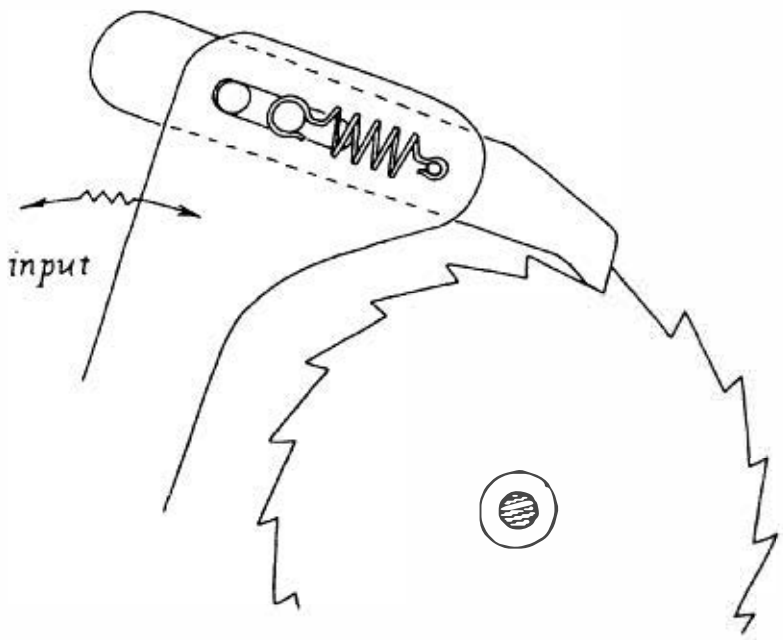


Fig. 4-11. Ratchet mechanism with a spring-loaded drive pawl. The spring would either reduce or increase impact stresses depending upon the rate at which the spring returned energy to the system and the rate at which the device was operating.

THE PROBLEM OF CONTROL

Another problem revealed by the spring-driven ratchet mechanism of Fig. 4–1 is the problem of control. When studying the motion of this ratchet at the beginning of the chapter it was seen that the collision between the ratchet and the ratchet wheel resulted in a series of impacts between the two bodies, with the ratchet wheel jumping ahead of the ratchet arm after each impact; the wheel was then slowed down by frictional drag until the ratchet arm could catch it. Ultimately, thanks to the energy loss during this process, the wheel and the arm moved together. Actually, a ratchet of this type functions more reliably when it has frictional drag in its bearing than it would if it were on a frictionless bearing. The amount of "jump" of the ratchet wheel after each blow is determined by a number of factors including the masses of the arm and wheel, the collision velocity, surface conditions, material spring rates, etc. It is possible to maximize or minimize the follow-through of the arm by using the proper inertia ratio between driver and driven links (and load), as will be seen in a moment.

But first, consider the potential difficulties of such a driver. It is always desirable in a machine for the driver(s) to determine the exact motions of the load(s). Perhaps "always" is too extreme, but certainly in most cases it is desirable. It is only on the golf links or in other game situations, where a driven member (ball) that follows an independent path of motion after being struck by the driver is tolerated. This state of affairs should definitely be avoided in most machines! Yet in the spring-driven ratchet this situation exists. It is only friction that (ultimately) makes the driven wheel stay with the driver. And friction can come when it is least wanted, then vanish when you are depending upon it. What is needed are intermittent motion devices in which the output is always under the control of the input; the position of the output link is determined by the position of the input link.

As will be seen, intermittent motion mechanisms vary widely in the amount of positional control they maintain over the driven members. Escapements control the output only at the beginning and end of motion—and then, only if they are properly designed. Ratchets require no-back pawls, non-overthrow pawls, and sometimes dampers to maintain even this much control. Interrupted gearing does a little better, but needs a locking ring to hold the output

during dwell periods. Genevas and star wheels also must incorporate locking rings. Clutches must be accompanied by brakes. Cams can generally take care of both motion and dwell periods, as far as load control is concerned; but again, only if they are properly designed.

Also, in almost every mechanism, there are moments during each operating cycle when the control situation gets a little difficult. Is the brake applied and the clutch then released, or vice versa? Does the locking ring become effective at the exact instant the drive pin leaves the slot in the Geneva? Does the stopping pallet of the escapement reach an interference position just before or just after the escape pallet has released the load, etc.? Because we are dealing with intermittent motion we are dealing with periodic change in the state of the mechanism; clutched–unclutched; motion–no motion; etc. And control during change is very difficult to achieve.

There is another factor that causes designers to be alert when dealing with intermittent motion. The magnified forces studied earlier distort the various links in a machine—elastic bodies—remember. The quasi-static model revealed that each link in a machine could be considered to consist of a rigid body connected to all the other links by springs. And as in Chapter 3, the motion of a mass-spring combination can be very different than the motion of the input member that excites it. How can the output of an intermittent device ever be made to behave?

Maintaining Control by Reducing Forces

One way to reduce the control problem is to minimize drive and stopping forces; using some of the techniques discussed under the third section in this chapter, "The Problem of Force Magnification." This will reduce the effectiveness of the quasi-static springs, resulting in dealing with bodies which appear to be "more rigid." Reducing forces will also reduce wear and, therefore, improve stability in general, as will be seen in the next chapter.

But we cannot use all of the force-reduction techniques and hope to improve control, simultaneously. An eddy current slip coupling must introduce positional uncertainty between driver and load in order to function. Feedback of some kind must be used, or the coupling must be overridden by brakes and other devices, to maintain full control. Dampers also reduce positional control while modifying the influence of the driver on the load.

The Impossibility of Full Control During Impact

Many intermittent motion mechanisms inherently involve impact; they require it, perhaps, to impart motion to the load. Examples include ratchets, escapements, interrupted gearing, some cam systems, some clutch-brake systems, etc. Can the designer avoid loss of control during impact by matching the load and driver inertia or the like? Table 3–1 (in Chapter 3) lists some of the methods of maximizing or minimizing the effects of impact. How can this discussion be related to the spring-driven ratchet control problem? To start, reconsider the linear collision of two elastic blocks, using the quasi-static model. As before, the blocks can be considered to be rigid bodies separated during the collision by a stiff linear spring. In Fig. 4–12A the block on the left, labeled M_1, is approaching the second block with velocity v_1. The second block at the beginning of the action is at rest, so that this is, indeed, a linear model of a ratchet wheel and drive-arm system. Block M_1, furthermore, is moving under the influence of a constant drive force, F.

To start with, assume that the masses of the two blocks are equal. The forces on each body are shown as a function of time in Fig. 4–12B. To begin with, there is a positive small force (F), on block M_1 and no force on block M_2. When the collision starts, the force between the blocks builds up sinusoidally as a function of time. This will be an equal and opposite force and will be negative in direction on M_1, and positive in direction on M_2.

As these forces develop in the bodies, there will be a change in velocity in each body, with M_1 tending to come to rest and M_2 starting to move. At the midpoint of the collision process, the velocity

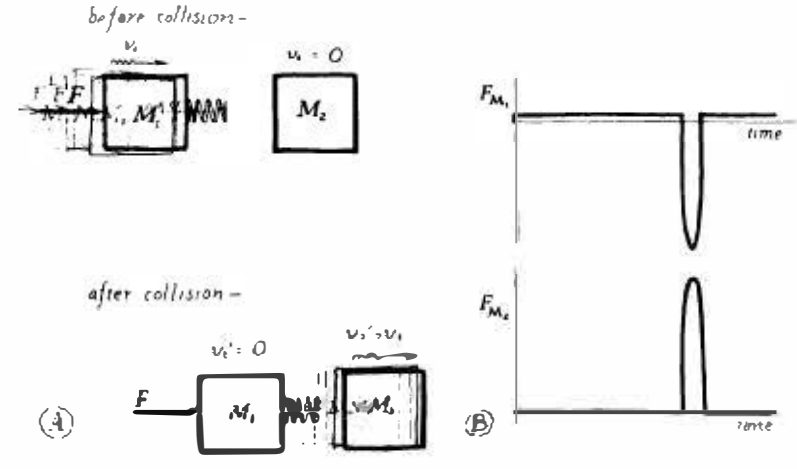


Fig. 4–12. Collision between equal masses. M_1 is the driver. In this situation it stops momentarily after the impact and the load (M_2) moves off with a velocity equal to the initial velocity of the driver. This happens even though there is a continuous drive force (F) on M_1. A very poor control situation.

of the two blocks will be equal to each other, and towards the right. This turns out to be the midpoint of the collision process, or the point at which the two bodies have reached maximum deformation during the collision: the spring, assumed to exist between the two bodies in the quasi-static model, is now fully loaded. If the collision process were to cease at this point, the two bodies would now travel together with a common velocity towards the right and there would be a single-impact collision. If we assume, however, that the bodies are elastic, then the spring between them will unload; at least in part. (In part, because there is energy loss during an actual collision and the spring will not return as much energy to the system as it has stored during the first half of the collision.) As the spring unloads its energy, it will push equally on the two blocks, pushing to the right against M_2 and to the left against M_1. These forces tend to increase the velocity of M_2 and decrease that of M_1. If the blocks are, indeed, equal in mass to start with, and if there is not too much energy loss during the collision process, then the unloading of the spring will produce just enough negative impulse (force × time) to bring block M_1 to a rest, and enough positive impulse to increase the velocity of M_2 to a value approaching the velocity of M_1 at the start of the collision.

It is not possible to determine from FAVD curves why the first block comes to rest and the second acquires the velocity of the first during a collision between bodies having equal mass, or no energy loss. In order to determine why this is so, it would be necessary to solve the equations for the conservation of energy and the conservation of linear momentum, simultaneously. However, for the purpose of acquiring a general feel for intermittent motion mechanics and mechanisms, it will be sufficient at this point, to state that this is indeed true. With equal masses and no energy loss, the first block imparts all of its energy to the second.

Relating this now, to the ratchet situation, if there is a ratchet arm and drive system striking a ratchet wheel and load of nearly equal inertia, then the chances are that the drive arm will come to rest upon striking the wheel and that the wheel will jump ahead with a velocity equal to the impact velocity of the drive arm. This is an efficient transfer of energy from driver to driven link, but probably, it will be difficult for the ratchet arm to keep up with and control the ratchet wheel. The arm will follow the wheel. It just will not follow closely enough to maintain control. This is, in fact, somewhat akin to the golf ball situation.

If the mass of the driver is increased substantially, and/or the mass of the load and driven link reduced, we find that the situation can be improved somewhat. The driven link will move away from the collision at some velocity greater than the collision velocity, also the driver does not change its own motion appreciably during the collision, but follows the load fairly closely. This allows the driver to control the load a little better, as seen in Fig. 4–13.

The larger the driver becomes in relationship to the load, the higher is the velocity imparted to the load during the collision. If the driver had an infinite mass, the load would be driven away at twice the velocity of the driver at the start of the collision as shown in Fig. 4–14. Their relative velocity is now V_1 (since the infinite mass would not slow down at all) and so the lack of control is the same as that of the equal mass system studied earlier. An infinitely large driver is, however, non-existent. Having a "fairly large" driver and a smaller load will usually tend to improve the control situation. It is not common, however, to find ratchets that are larger than their loads!

If the driver is small, compared to the load, then it will rebound or reverse its velocity during the collision and the load will start very slowly. In other words, the load's initial velocity will be significantly less than that of the velocity of the driver, upon

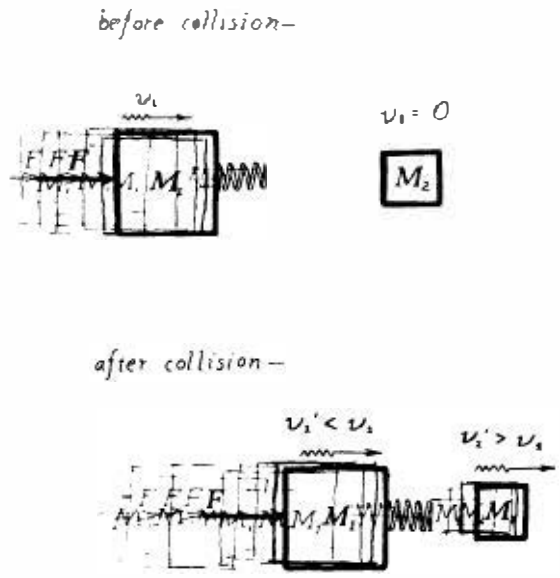


Fig. 4–13. Large drivemass striking small load. The driver continues to move following impact, though with reduced velocity. This is a better control situation than that shown in Fig. 4–12, and may be the best of all possible choices.

impact. If there is a drive force on the driver, as is common in machines, its reverse will reverse again, and eventually the driver will come back to take another shot at the load; but, in general, this is a very bad arrangement and it takes a long time for driver and load to get together. Control of the load is difficult or even impossible for the driver. (See Fig. 4–15.)

Although the three cases: a very small driver, an equal (to the load) driver, and a very large driver have been discussed, in none of them will the load and driver stay together during and after the first impact. Unfortunately, there are no conditions between these points where the bodies will move together after the impact. Only in a fully plastic collision (all the energy absorbed by the bodies during collision is dissipated within the bodies) will they move together after the impact. In such a collision, however, the bodies are deformed and stay deformed, which obviously cannot be tolerated in an actual machine design situation. Inevitable, therefore, is the conclusion that whenever impact occurs in a machine there is always a control problem to be faced. The driver and load will not, and cannot, stay together until after a considerable amount of energy has been dissipated in the machine by multiple impacts.

The only way out of this dilemma is to try to

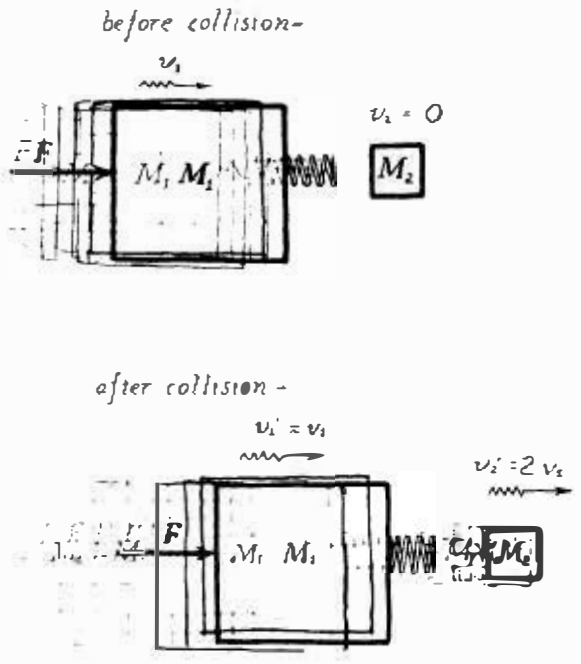


Fig. 4–14. Very large driver striking a very small load. The driver velocity is not affected significantly by the impact, but the load moves off at twice the driver velocity after being struck; thus this system produces the same lack of control as the system of Fig. 4–12.

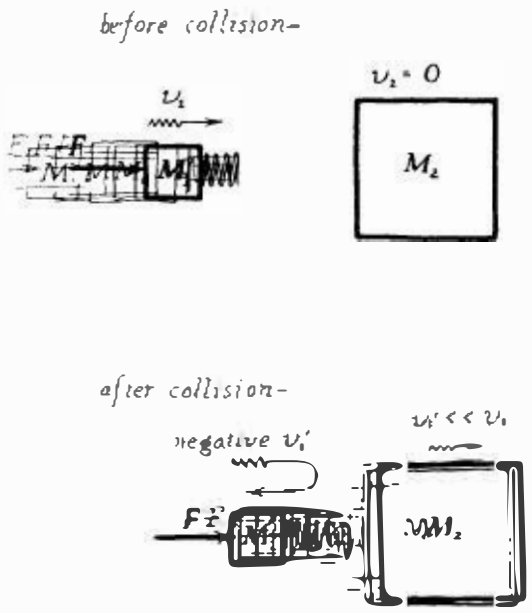


Fig. 4–15. Small driver striking a large load. The driver reverses direction of motion on impact. A very poor control situation.

design a mechanism that avoids impact; or failing that, to try to design one which reduces the magnitude of impacts. Don't "load the spring" and you will retain better control.

THE PROBLEM OF SPEED

The forces generated during impacts and during sudden changes of velocity in intermittent motion mechanisms eventually limit their operating speed. Remembering the quasi-static model, we see that the mechanism shown in Fig. 4–16, part (a) can be represented by the series of rigid bodies and springs shown at the bottom of that same illustration, part (b). The motor shaft on the left is trying to drive the four-bar linkage, which drives the ratchet pawl, which indexes the ratchet wheel. Because of impacts between the ratchet pawl and wheel, very high forces are generated at that point. These forces are high enough to distort not only the ratchet wheel and pawl, but any other links between that point and the original power source in most designs. The situation is made considerably worse if there is any backlash or clearance in the joints of the four-bar linkage; as there almost always is.

Any success in driving the ratchet wheel will depend upon the impact-produced distortions and vibrations that modify the intended geometry of the elements of the machine. Remember, too, that force is proportional to acceleration which is the rate-of-change of velocity. And velocity is the rate-of-change

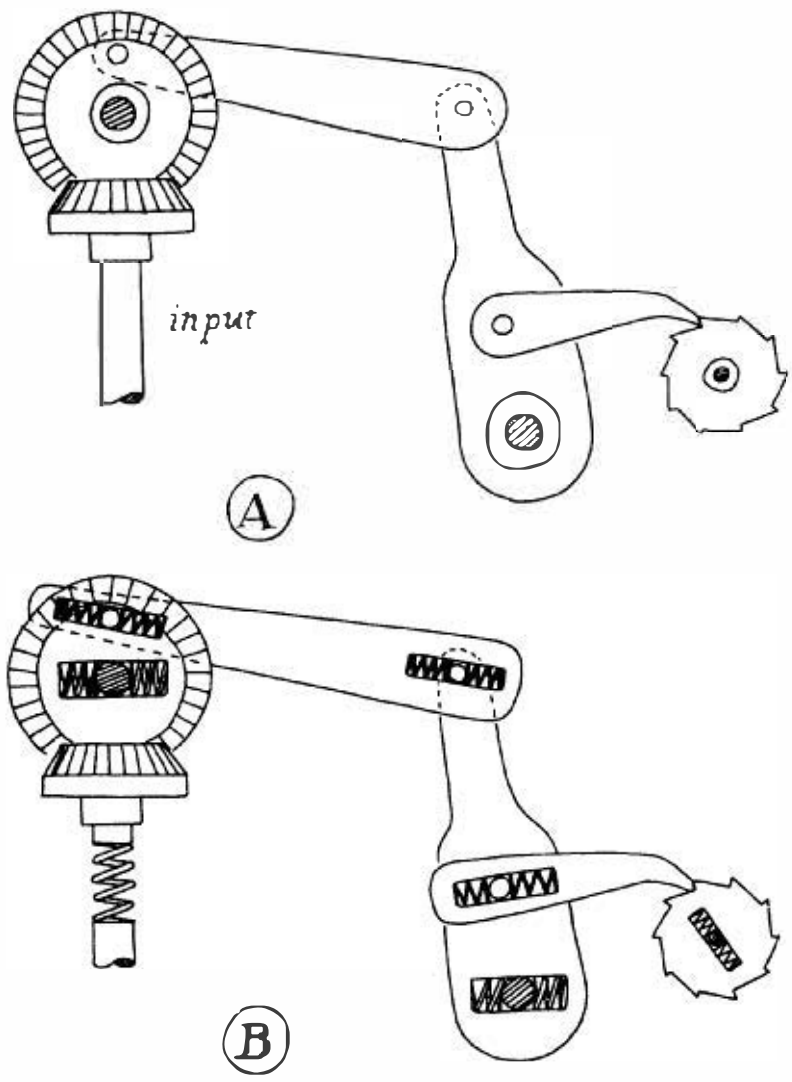


Fig. 4-16. (A) Four-bar linkage ratchet mechanism. If this mechanism is driven at too high an indexing rate its performance will become elastic and it will behave just as the quasi-static model shown in (B), at the bottom of the illustration. A designer's nightmare!

of displacement. As the operating speed of the machine increases, it requires greater and greater forces to make the various links move through their intended displacement cycles. Drive torques must be increased drastically as the desired speed is increased. As with impact, this results in increased deformation and vibration within the machine. At some input speed, galloping or hesitation would begin to be apparent in the ratchet wheel. The connecting links have become too rubbery, or jammed, to drive the wheel consistently. At a higher speed no motion at all might be seen in the ratchet wheel. Of course, at speeds of this magnitude, the stresses could be large enough to destroy the machine, or the motions of various elements could get so far "out-of-time" that destructive jamming could occur.

Again, we are faced with a situation that will never occur in the world of "rigid body" mechanics where geometry is king, and there is no limit on the theoretical speed at which a four-bar linkage can be put through its paces. But in real life there is a limit, as explained by our quasi-static model and by our study of the vibration of coupled oscillators. The closely related problems of control, force amplification, and speed limitation are very real in the land of intermittent motion.

TORQUE CHARACTERISTICS OF DRIVERS

Most intermittent motion mechanisms must be driven by something else in the system. Input must be provided from a rotating shaft, an electrical motor, an hydraulic motor, a solenoid, a spring, or some other type of power source. Perhaps the only exception would be the stepping motor which is self-powered and which is discussed as an intermittent motion "mechanism" in Chapter 14 of this book. A motor is not really a mechanism, of course, but steppers are such an important way of providing intermittent motion in this day and age that we could not neglect them.

In any event, most of the devices we shall consider will require a driver of some sort; and a few moments should be spent at this time in considering some of the characteristics of typical drive systems, since they will inevitably influence the performance of our mechanisms.

Very commonly we will be driving our mechanisms from a constant velocity source, usually a rotating shaft whose velocity does not change significantly as long as the machine is turned on. Such a shaft, in turn, could be driven by a synchronous motor or by a variable-speed motor whose torque capability is such that the loads imposed on it by the intermittent motion mechanism do not affect its overall velocity. In most cases, we will assume that the constant velocity shaft can also provide considerably more torque than is required. And again, this is not a bad assumption in most machine design situations (as long as we have selected our power source properly). Most of the motion curves you will find in this text are based on the following type of drive input: constant velocity with more than adequate torque. Whatever happens in the output of the intermittent motion device the input will keep turning around at the same speed.

In some situations, however, torque is definitely limited, even though the input runs at a constant velocity. Many systems, for example, include a slip clutch of some sort that will limit the peak torque

which can be delivered through an intermittent motion mechanisms to the load. If the clutch were to slip at too low a torque level, the mechanism might not be able to produce the peak accelerations required by its output motion curves. If this were the case, the machine and mechanism might malfunction, and the designer must be sure to avoid this. One example of the use of a slip clutch in an intermittent motion system is given in the design example in Chapter 16, where it is used to provide manual control over an intermittent motion rather than to limit torque. But it will limit torque, too, and the designer must take this into account.

Other types of drivers will run at a velocity that is proportional to the load on the driver. A spring, for example, falls in this category. It is definitely torque limited, but will not slip and so will produce output acceleration proportional to the load. We will see examples of spring drivers in our studies of ratchets and inverse escapements as two examples. Here the output motion curves are strongly influenced by the torque capability of the driver.

Some electrical motors will also run at a velocity that is proportional to the load on the system. Again, the designer must take this factor into account when constructing motion curves for the output. Check your motor supplier for the speed-torque characteristics of his products.

Finally, some drivers will produce torque which is a function of time. Electrical solenoids, for example, produce torque in proportion to the electricity passing through their coils. Since it takes time for this electricity to reach the peak value, it takes time for the solenoid to develop full torque. Again, this sort of driver is frequently used with ratchets and inverse escapements.

Calculating Drive Torque Requirements

As mentioned above, the motion curves you will find throughout this text describe the motion of the *output* of various intermittent mechanisms. These curves usually represent output displacement, velocity and acceleration as functions of time or of input displacement. By multiplying the output acceleration by the inertia of the load, we could, of course, calculate the torque required to accelerate the load in accordance with that acceleration curve; and if we are not careful, we might assume that the resulting torque curve would define the torque which the input driver must produce to move the load, but

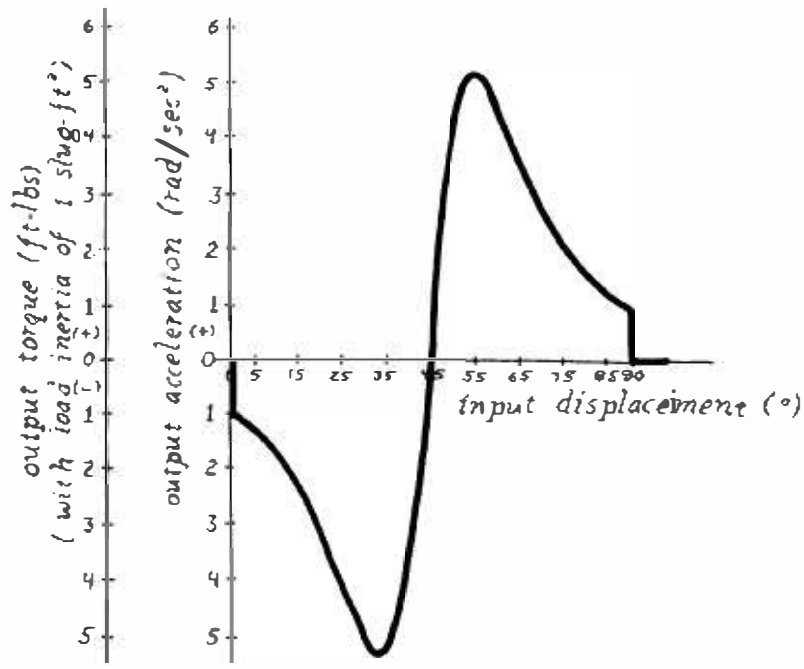


Fig. 4-17. Acceleration of the output member of a four-slot Geneva mechanism as a function of input displacement when the input is rotating at a constant angular velocity of one radian per second. The same curve also shows the torque the driver must exert on the output member to produce this acceleration pattern in a load having an inertia of one slug-ft². (See the second vertical scale.)

this is not the case. As we saw in Figs. 1-9 through 1-12, the torque produced on a load is not necessarily equal to the torque required of the driver. In fact, it is probably rare that we will find a situation in which input and output torques are equal to each other. Let us consider an example to be sure we understand this very important point.

Figure 4-17 shows an acceleration curve for the output of a 4-slot Geneva mechanism that is rotating in a counterclockwise direction with an input velocity of 1 rad/sec. (See Chapter 9 for a complete discussion of Genevas). Notice that the peak acceleration produced by this mechanism has a magnitude of approximately 5.2 rad/sec². (The acceleration is negative because rotation is counterclockwise.) If we want this Geneva to drive a load having an inertia of 1 slug/ft² (or 1 ft-lb-sec²) then there will be a one-to-one relationship between the acceleration and torque scales. An acceleration of 5.2 rad/sec² will require a torque on the load of 5.2 ft-lbs. Let us make this load assumption, therefore, and add a torque scale to the illustration, as shown. We now have a graph of the acceleration of the output member of the Geneva—and of the torque exerted on the output by the input to achieve this acceleration.

We have defined the amount of torque required to drive the load. What drive torque must the input member now provide to produce this amount of

output torque? To answer this question, we must make geometrical studies of the Geneva mechanism.

Figure 4–18 shows the 4-slot Geneva after the input has completed five degrees of motion. The graph in Fig. 4–17 would indicate that the load torque requirement at 5 degrees is 1.2 ft-lbs. With the aid of the drawing we can determine the line of action of the force transmitted from the driver to the output member of the Geneva. We are assuming, in this analysis, a frictionless system to clarify the calculations and are following, therefore, the example illustrated in Fig. 1–10. The force between input and output must be perpendicular to the surface of the drive slot. In "real life" we would undoubtedly have to assume that friction were present to some extent, would have to estimate its magnitude, and would follow the procedure illustrated in Figs. 1–11 and 1–12; but for the moment, we assume no friction.

To return to Fig. 4–18: as mentioned above, the drawing shows the line of action of the force exerted by the driver on the output member (or vice versa). Once this line of action is known, we can draw perpendiculars from it to the centers of rotation of input and output members, respectively. We know from Fig. 4–17 that we require a torque of 1.2 ft-lbs in the output at this point (5 degrees of input motion) in the cycle. The force required to produce this amount of torque at a radius of 1.8 ft is found by making a simple calculation:

$$F = \tau / R_1$$

$$F = 1.2/1.8 = 0.67 \text{ lbs}$$

The force, therefore, is 0.67 lbs. The reaction force (force exerted by the output on the input) will also be 0.67 lbs in the opposite direction. The drawing shows that the line of action of this equal and opposite force will pass 0.2 ft away from the center of rotation of the input member or driver. The drive torque, therefore, after 5 degrees of input motion will be:

$$\tau_2 = FR_2$$

$$\tau_2 = 0.67 \times 0.2 = 0.134 \text{ ft-lbs}$$

Since both torques act in a direction to cause counterclockwise motion, they are both negative. The driver, however, continues to turn in a clockwise direction under the greater influence of the drive motor.

At this point in the cycle, then, an input torque of 0.134 ft-lbs is producing 1.2 ft-lbs of torque on the load. At first glance this seems difficult to believe, but I am sure that a study of the geometry of the

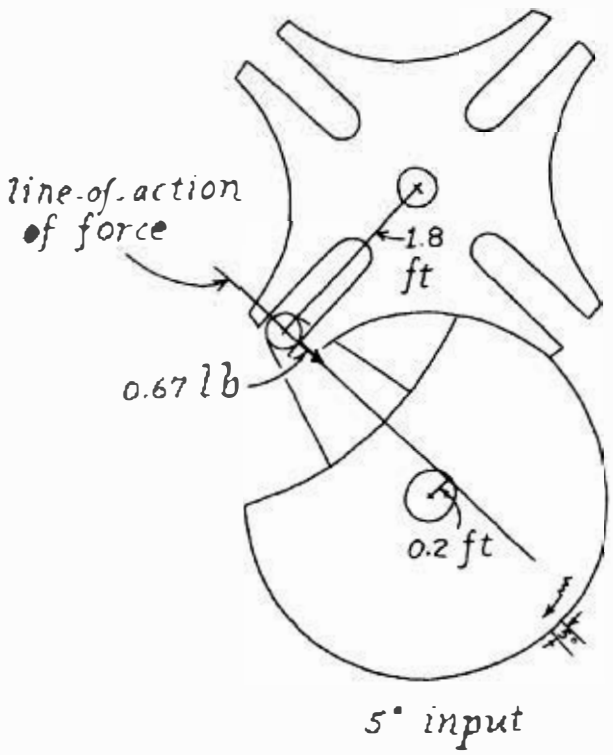


Fig. 4–18. Construction for determining the force generated by the input member on the output member (and vice versa) after five degrees of rotation of the four-slot Geneva. Input and output "lever arms" for this force are also shown. Resulting input and output torques are noted. They are both negative because both are counterclockwise

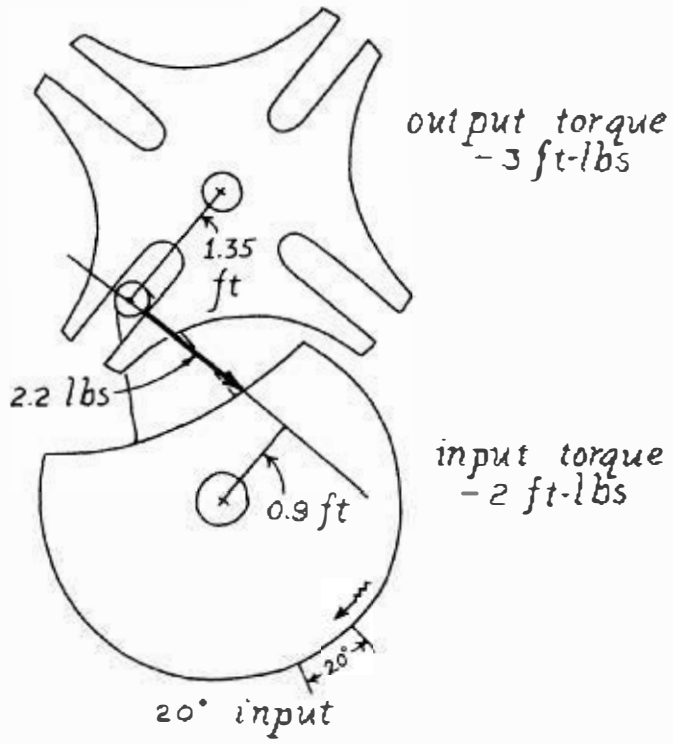


Fig. 4–19. A force and torque construction for the same four-slot Geneva after 20 degrees of input motion.

mechanism will convince you that this is indeed the case. If you need additional conviction, look at the velocity curve for a four-slot Geneva (this curve is given in Chapter 9). You will see that the velocity of the output is nearly zero at the instant shown in Fig. 4–18. The input, however, is turning at a constant velocity of 1 rad/sec in our present example. This means that we have a significant velocity ratio or "gear ratio" between input and output members at the 5 degree point and would expect, therefore, to find a large difference between input and output torques.

This procedure is repeated in Figs. 4–19 and 4–20, each one representing a different instant of time during the motion cycle of our Geneva. In Fig. 4–19, for example, the input has moved 20 degrees, the output requires 3 ft-lbs, and the input torque is 2 ft-lbs. In Fig. 4–20, input and output torques are equal. There are only two instances during each cycle of the Geneva when this is true; one during the acceleration phase and one (that shown in Fig. 4–20) during the deceleration phase (again, remember our curves are "upside down" because torque is counterclockwise).

If the input torque requirements were calculated in this way for six or eight different positions of the Geneva during one cycle, we would have enough information to draw a complete input torque curve.

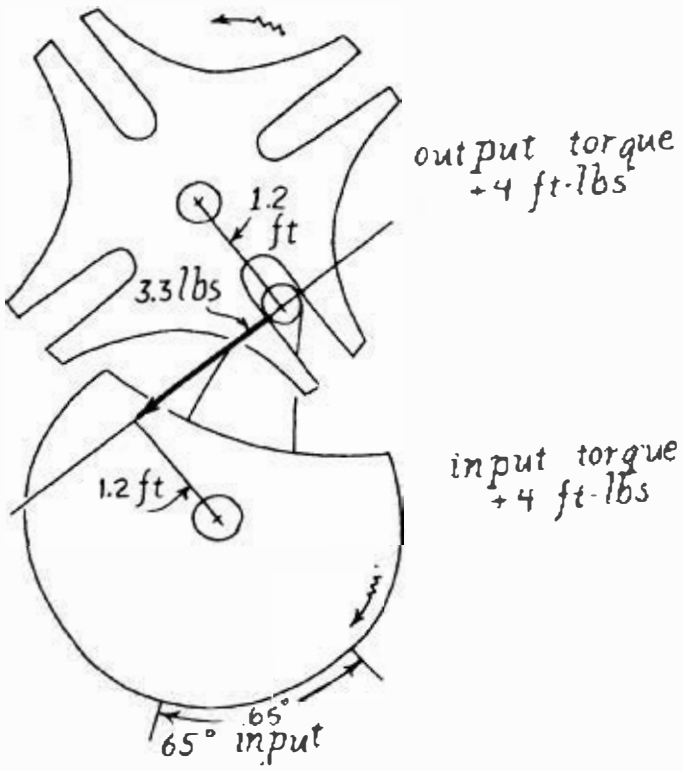


Fig. 4–20. Force and torque calculation for the four-slot Geneva after 65 degrees of input motion.

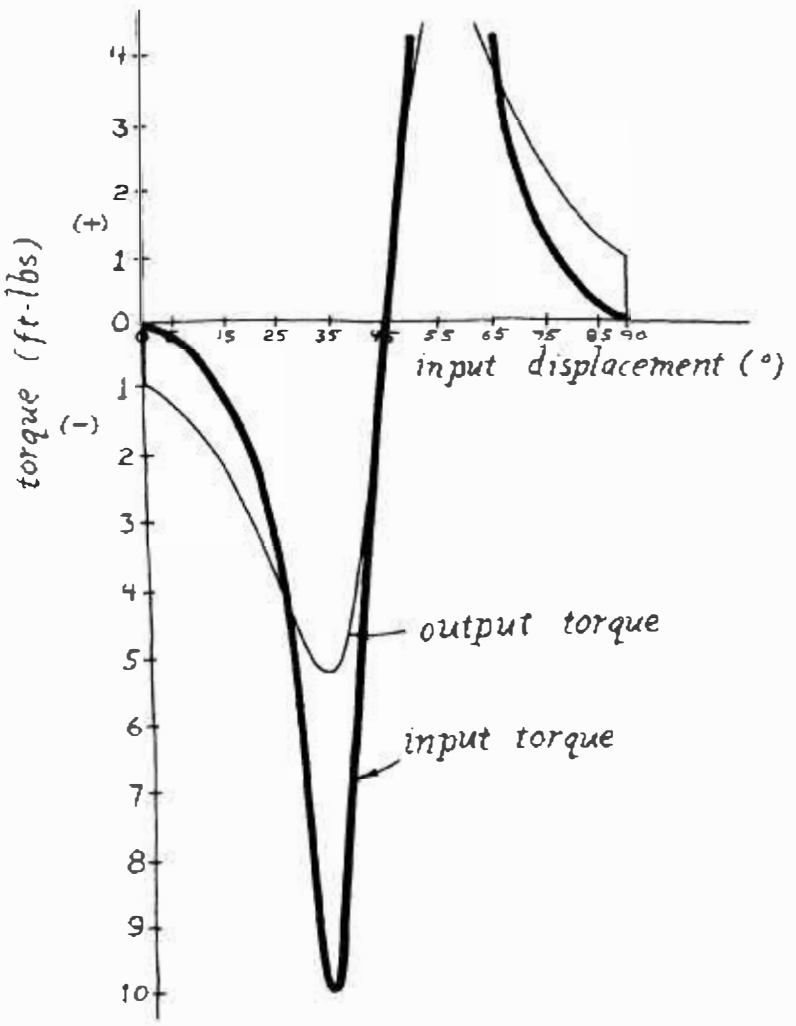


Fig. 4–21. Complete torque curves for the four-slot Geneva for one motion cycle. "Input torque" is the torque exerted on the input by the output member; "output torque" is that exerted on the output, or load, by the input. The two curves are similar in shape, but differ considerably in magnitude. Again, the horizontal axis represents displacement of the input, and output torque requirements are based on a constant input velocity of one radian per second.

This is shown in Fig. 4–21, superimposed on the original output torque curve. You will note that the two curves are similar in shape, but that the peak input torque is nearly double the peak output torque for this particular mechanism. This could be a significant design disadvantage in many situations. Cam designers, especially, worry about input torque curves and try to find cam profiles that will minimize peak input torque requirements.

In some other mechanisms, load-and-fire escapements, for example, input and output torque curves differ drastically in both magnitude and shape. Therefore, in most design situations it is best to consider driver characteristics and requirements before calling your intermittent mechanism design completed.

CHAPTER 5

Stability and Wear in Intermittent Motion Mechanisms

Types of Wear

The service life of any machine is determined by its stability in use. As long as the shapes of the various parts of the machine are not altered too much by wear or deformation, the principal parts of the machine do not break, and the various loads experienced by the machine (including the friction load), as well as the input power to the machine remain stable, the machine will continue to function as designed. Unfortunately, all operating machines experience gradual changes in all of these factors. The geometry of parts changes as they wear. Loads frequently increase as wear particles accumulate and increase frictional resistance to motion. Drivers (solenoids, motors, etc.) decay as they age magnetically, as brushes and bearings wear, as insulation evaporates, etc. The designer cannot prevent wear and the ultimate "death" of his machine; but by understanding the various types of wear, by being able to predict where it will occur, and by designing the various parts of his machine so the whole will continue to function correctly even when worn, he can extend the life of his machine to some degree.

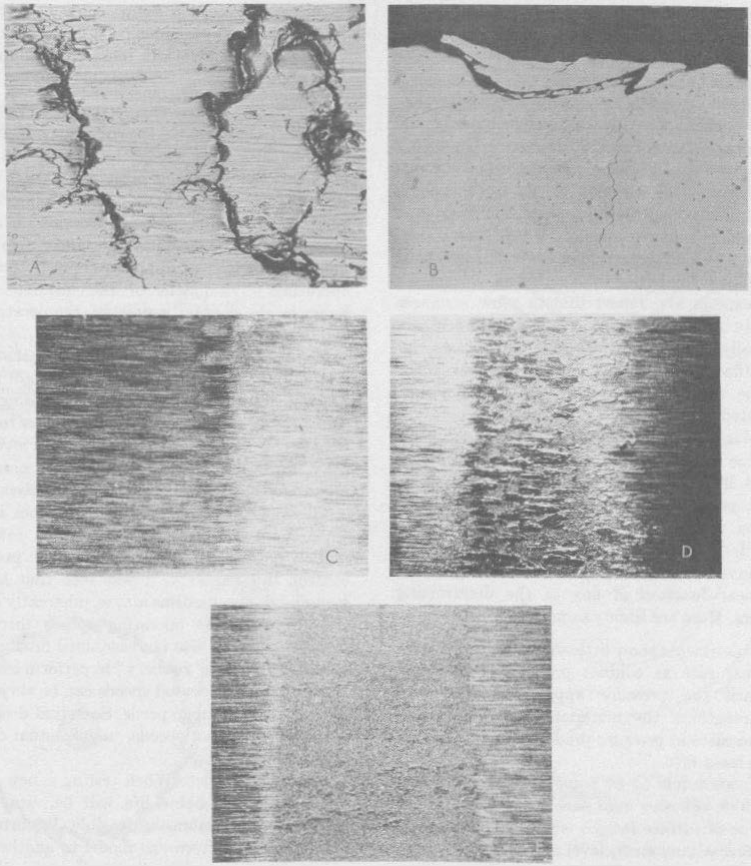
Intermittent motion mechanisms wear just as everything else, and are perhaps, as a class, more affected by wear than other types of machine devices since they are exercises-in-instability to begin with.

Experts differ somewhat in defining and classifying types of wear. But, generally speaking, four types are identified:

1. Plastic flow or ploughing, where the surfaces of one or both of the bodies in contact (sliding contact) deform, but particles are not immediately removed.
2. Adhesion or transfer wear, where particles from one body in sliding contact with another, adhere to the other body and are wrenched free from the first. Sometimes, such particles remain on the second body, or they can fall free as wear particles. At times they later transfer back to the first body.
3. Surface fatigue caused by cyclic stresses produced in surfaces as bodies periodically impact or roll across each other.
4. Abrasive wear or fretting, a grinding action in which wear particles resulting from other types of wear act as abrasives to permit one mating part to grind away the other.

These are the four mechanical types of wear. There are also chemical and corrosive types which are factors in the ultimate death of many machines, but these are not commonly cured by changes in the configuration of the design and thus are not pertinent to this discussion.

The wear in a given design can usually be classified as adhesive, abrasive, etc., only by microscopic examination. And then only by people who know what they are doing. Figure 5–1 shows some microphotographs of different types of wear. Figures 5–1A and 5–1B show surface cracking. The first photo-



Photographs courtesy of Massachusetts Materials Research Inc.

Fig. 5–1. Microphotographs of worn metallic surfaces; (A) surface cracking; (B) surface cracking; (C) [illegible]ar; (D) adhesive wear; (E) [illegible]

graph is a plastic replica of the worn surface; incidentally, an interesting new process in which a sheet of softened acetate is used to record the features of a surface. Figure 5–1C shows abrasive wear; Figs. 5–1D and 5–1E show adhesive wear.

[illegible]

In general, the rate at which parts wear will be reduced if the parts are

1. Made harder by chemical or heat treatment
2. Operated at lower stress levels
3. Operated at lower speeds
4. Operated at lower temperatures
5. [illegible]
6. Are properly lubricated

Watch out for generalities of this kind when dealing with wear, however. There are many exceptions to the rule. For example, my company discovered

years ago that interrupted gears made of plastic would outlast steel or brass gears in a given counter design. The plastic was much "softer" than the metal, yet gave much longer service in the same design. There are probably several reasons for this. First: the plastic must distort under impact loads and so act to dampen shocks. Second: the impacting parts have lower mass, thus reducing impulse. Both factors reduce peak stresses. Third: as the plastic distorts it brings larger surface areas into contact at load points, reducing contact pressures. The result: the softer part lasts longer than the hard! But hard steel usually wears better than soft steel.

Wear experts also report that in some instances an increase in sliding velocity or contact temperature can actually reduce wear rates; in other cases, increasing these factors can drastically increase wear. And some materials will wear less rapidly when unlubricated.

One thing, however, is certain, that the wear rates of machine parts will be hard to predict. Small variations in such things as sliding velocity, operating temperature, alloy content, surface finish, lubrication, etc., can have a drastic, though mysterious, effect on the life and performance of machine parts. One reason for this is that wear rate is usually not a linear function of any of the determining parameters. Here are some examples:

1. Experiments show little variation in adhesive wear rate as contact pressure is increased until the pressure approaches the yield strength of the material. Then very small increases in pressure produce drastic changes in wear rate.
2. There seems to be some stress limit below which adhesive wear *never* occurs. This is not true of surface fatigue which will eventually occur at any stress level.
3. Surface fatigue, however, usually produces no visible wear particles, or the like, for many thousands or even millions of operating cycles. Then the surfaces will start to deteriorate very rapidly. When stress levels are low, however, this delay in the onset of visible wear can result in what amounts to zero wear for the life of the product. (See Fig. 5–2.)
4. Some engineers feel that surface-fatigue wear rates are proportional to the ninth power of applied stress. This means that if the load at a given wear point doubles, the fatigue wear rate will increase by a factor of 512! Or, more realistically, a ten percent increase in stress level can more than double fatigue wear rate.

One result of all of this is that it is virtually impossible to determine the rate at which a given machine will wear except by actual tests of that machine. Bench-testing sample materials can show RELATIVE wear rates of different material combinations, but cannot yield information on the actual wear rates of any machine using those combinations, unless the bench test duplicates exactly the complete operating environment of the ultimate machine: stress levels, contact geometry, temperature, lubrication, contaminants, etc.

For the same reason, wear tests cannot be accelerated. Remember that wear rates are not linear functions of stress, temperature, or sliding velocity. Accelerating a machine, running it faster to speed up wear tests, will usually increase stress levels drastically. Parts that might never wear in normal operation may now go to pieces very rapidly.

There is another factor that becomes important when accelerated tests are made on intermittent motion devices. In the section on the problems of control, in Chapter 4, it was seen that high-speed stop-and-go mechanisms almost inherently have control problems. As operating speeds increase, the control problems also increase, until finally, the machine becomes too "rubbery" to perform as intended. Wear modes at elevated speeds can be very different from those at design speeds. So actual designs must be tested at actual speeds, under actual conditions to get valid results.

A word of advice: When testing a new design to see what its probable life will be, test as many models as is economically feasible. Performance can vary drastically from one model to another because of the factors already discussed, even though the models appear to be geometrically identical. This nonuniformity of result is a very common problem with intermittent motion mechanisms and to overlook this factor can prove to be costly.

Wear in Intermittent Motion Mechanisms

The potential wear points in intermittent motion mechanisms can usually be determined by inspection if the concept of degrees of freedom is kept in mind. Any machine body can translate in three directions, or rotate in three directions (or a combination of

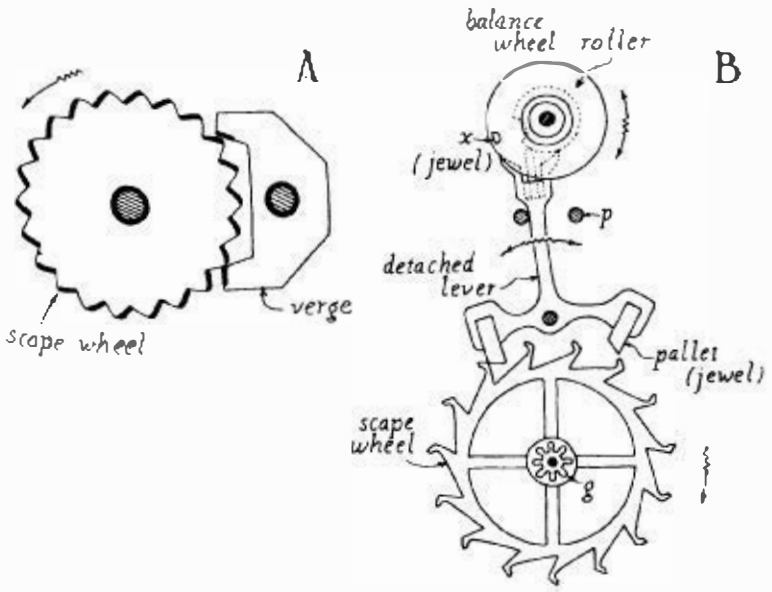


Fig. 5–2. Typical wear points in two timer escapements. Wear points in the "runaway" escapement (*Left*), are indicated by heavy lines. This particular escapement is an "untuned" escapement (see Chapter 11), and is subject to high impact stress. By contrast, the "tuned" escapement (*Right*), will run more than a lifetime if kept clean and oiled. Here the principal impacts occur between jewels (ruby or sapphire) and highly polished or burnished hardened steel. Some tuned chronometer escapements have been operating for over 100 years with no signs of wear.

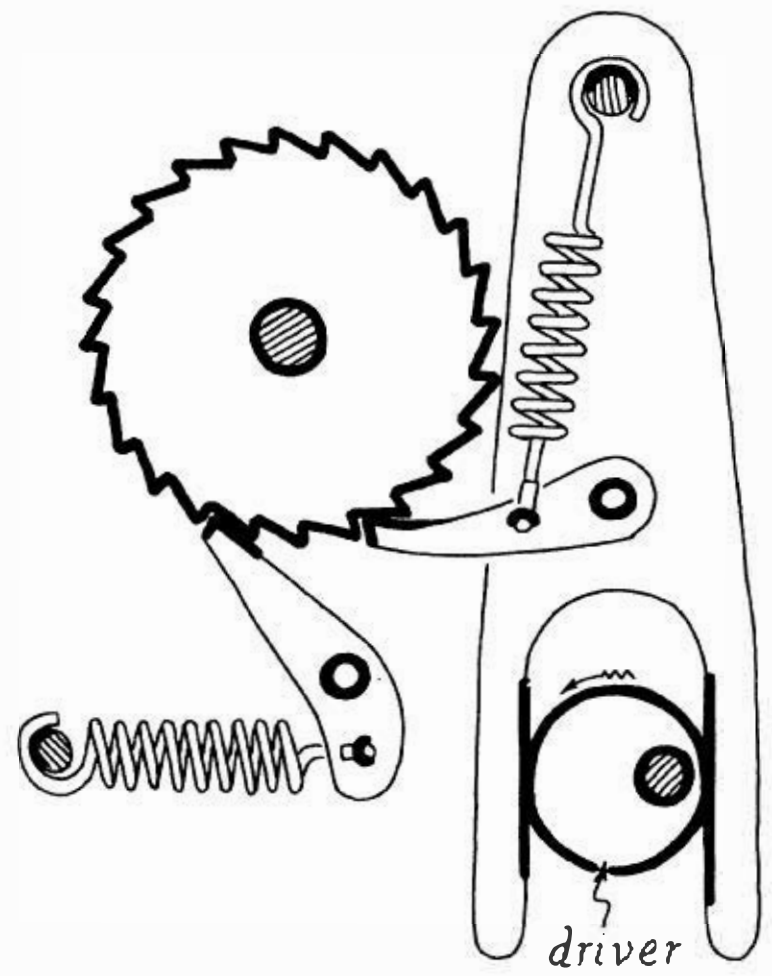


Fig. 5–3. Typical wear points in a cam-driven ratchet driven mechanism. Again, wear points are shown by heavy lines.

these six), unless constrained by shafts, bearings, slides, etc. (See Figs. 1–14 through 1–16.) Wear points always occur at the points at which the body contacts such constraints, as well as at the points where the input forces are applied to the body, and the points at which the body exerts force on its load. Typical wear surfaces for three intermittent mechanisms are shown as heavy black lines in Figs. 5–2 through 5–4. In contrast with the untuned mechanism shown in Fig. 5–2A, Fig. 5–2B shows a "Tuned

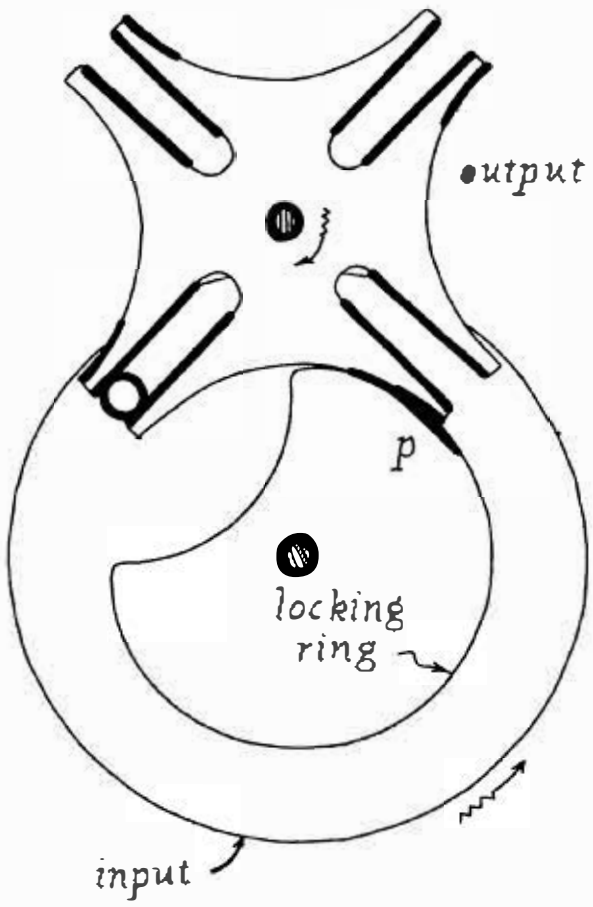


Fig. 5–4. Heavy black lines indicate the typical wear points in a Geneva mechanism.

Escapement" which, if kept cleaned and well oiled, will have a long life expectancy.

Intermittent motion devices are also subjected to high stress levels because of the forces generated during impact or during sudden changes in velocity. These forces, however, are usually of very short duration and scientists have found that materials can withstand high forces more readily for short periods of time than they can for long periods. This is not to say that intermittent motion devices can tolerate high stress levels indefinitely without wearing, just because such stresses are of short duration, but it does say that they sometimes operate at stress levels that would wear out continuous mechanisms

very rapidly. As a result, they sometimes exhibit life characteristics that textbook and handbook data would indicate is impossible. Most books, however, have not really come to grips with the world of intermittent motion.

Most intermittent motion mechanisms are subjected to cyclic stresses because impacts or cyclic rolling occurs someplace in the mechanism train. As a result, such mechanisms experience surface-fatigue wear more commonly, perhaps, than do other types of mechanisms.

As suggested earlier, surface-fatigue wear differs from other types of wear in several ways. For example, it was learned that fatigue wear rates may be proportional to the ninth power of stress levels. If this were true, a ten percent reduction in operating stress level would increase the life of a mechanism part by 258 percent. A 30 percent reduction would increase the life by 2,500 percent. By comparison, the stress wear-rate relationship of abrasive wear shows that a 50 percent reduction in stress level will only double the life of the mechanism part.

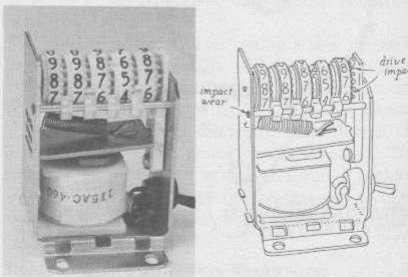
These studies of the performance of intermittent motion showed us that stress levels are extremely difficult to control where impact is involved. If the duration of an impact is decreased from 10 milliseconds to 9 milliseconds, the stress level in the surfaces is increased by 10 percent. This means that life will be reduced by a factor of 2.58. Little differences make a *lot* of difference in performance and life. This is a frequent theme when dealing with intermittent motion.

Surface-fatigue wear rates, like some other types of wear, also vary exponentially with operating temperature. Mentioned previously, it is known that surface fatigue characteristically produces little wear for a long period and then produces rapid failure.

All of these factors make the life of an intermittent motion device extremely difficult to predict. It also makes the life of such a device extremely difficult to determine experimentally. One of the characteristics of this class of mechanisms is that they exhibit a great difference in life from one model of a given design to another model of the same design. It is not at all unusual to get a variation in lifetimes of 50- or 100-to-1 in a large sample of a given design. What appear to be small differences in the shapes of material compositions, driver forces, backlash and pre-travel, drive voltage, operating temperatures, inertias and densities of parts, etc., result in small changes in stress level which are magnified by the nonlinearity of surface fatigue to produce large differences in wear rates.

It should be mentioned, too, that surface-fatigue wear is found not only at those points in the machine where impact has been designed in; for example, in a ratchet-wheel tooth or drive pawl, but at all other points in a machine, that are connected to an impacting or cyclic-rolling pair. As noted in the coupled oscillator model for an elastic-body machine, we realize that any sudden disturbance at one point in the machine will be transmitted as a serious of shocks and vibrations throughout the machine. Therefore, surface-fatigue wear occurs in the bearings of the ratchet wheel just as it does in the ratchet teeth; and even more mysteriously, surface fatigue will appear in the bearing surfaces in the frame that supports the shaft on which the ratchet wheel is mounted. Generally speaking, wear will be most rapid at those points in the mechanism train where clearances and/or backlash are greatest. This can sometimes result in some very unpredictable failures.

The counter illustrated in Fig. 5–5, for example, usually wears most rapidly at the input end, where a solenoid-driven inverse escapement drives the first count wheel. Here, where there are repeated, direct blows of the driver on the load, we would expect this to be the ultimate wear point, and it usually is. But I recall one test in which one counter failed when the long shaft on which the transfer pinions (intermittent gears between count wheels) are mounted, became so loose at the far end (most



Photograph courtesy of the Veeder-Root Company

Fig. 5–5. Wear occurs at the point of greatest clearance in many situations, not always at the expected point. In a counter shown in photo (*Left*) and detailed drawing (*Right*), for example, wear can sometimes occur on the left-hand end of the pinion shaft even though the drive impacts are introduced on

remote from the driver), that the pinions jammed the count wheels. The pinion shaft in this design is normally fixed to the frame. It is not intended to rotate. However, the pinion shaft was slightly loose in the frame on the particular counter tested. This small clearance allowed the shaft to impact the frame every time a load was put on the first or right-hand pinion. The repeated impacts gradually opened an ever larger hole in the frame until the shaft misalignment became so great that the mechanism was finally jammed.

Figure 5–6 also illustrates the fact that shocks and impacts are transmitted throughout an entire machine and produce unexpected wear phenomena. The illustration shows two systems used at Veeder-Root for life testing small ratchet-driven counters. In system (A), a four-bar linkage is used to drive the counter; the input motion, therefore, is roughly sinusoidal. At (B), a large star wheel periodically strikes the input lever to the counter, driving it in that way. Here the input is a series of blows. The counters being tested by both machines are identical; both are full of intermittent motion devices (ratchet inputs, mutilated pinion transfer mechanisms, etc.). But the life test results obtained on these two testing machines differ by a factor of nearly 30-to-1. Counters driven by the four-bar crank system operate much longer than those driven by the star wheel. And the modes of failure do not differ much in the two tests. The same failures occur in the same places in each, but at greatly different rates.

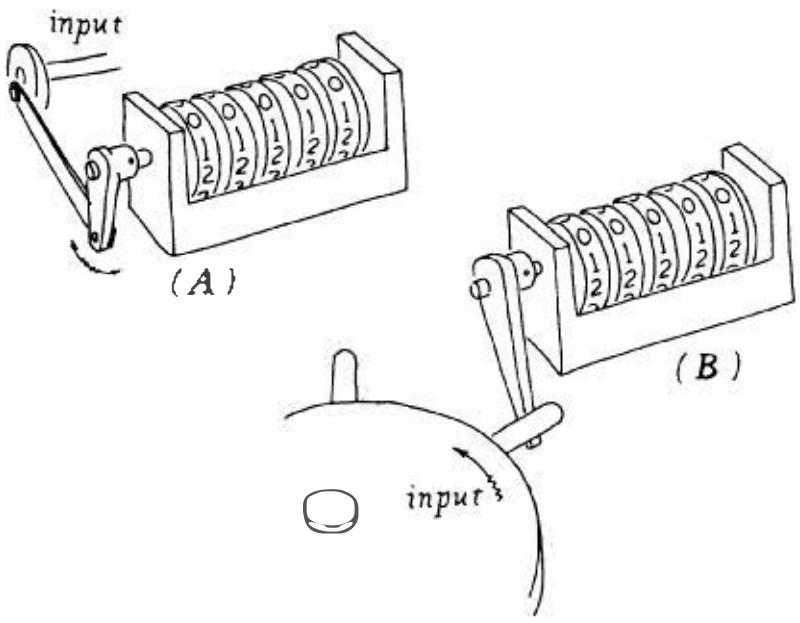


Fig. 5–6. Impact at one point in the machine will be transmitted throughout a machine and can cause wear at a number of different points. Typically, the useful life of a counter will be ten times greater when driven by the crank mechanism of (A); than when driven by the spoke wheel (impact) system, shown in (B).

Counters driven by electrical stepping motors also last much longer than the same counters driven by impacting mechanisms. Stepping motors, as will be seen in a later chapter, involve no impacts or metal-to-metal contact of any kind. Counter life can be increased by a factor of 100, or more, by switching to this sort of drive.

Another thing which is characteristic of intermittent motion devices is that their functional life is often strongly dependent upon the cyclic rate at which they are operated. By cyclic rate is meant the number of cycles per minute; not velocity. In a ratchet mechanism, for example, the drive forces and load forces (and, therefore, the impact and operating velocities) are the same in a given cycle whether or not the machine is being operated at a rate of 10 cycles per minute, or is only operated once a day. Why, then, should the ratchet mechanism fail to function properly after only 1,000,000 cycles when operated at a high cyclic rate, when it will go 200,000,000 cycles when operated at a lower rate? Measurements of wear, furthermore, show that wear rates per cycle are the same in both cases, as would be expected. Yet almost always, a strong correlation between performing life and operating cycle rates is found.

The answer lies in the fact that intermittent motion devices are always struggling on the verge of instability. Internal vibrations, distortions, discontinuities, etc., can limit the operating speed even if force magnification does not produce destructive stresses. Although identical ratchets operated at two different speeds can exhibit identical wear rates, the machine being driven at the higher speed will be more sensitive to small amounts of wear; and its performance, therefore, will be degraded more rapidly than will that of the machine operating at the lower speed. At some speeds the vibrations and distortions caused by a given impact will not have "settled out" before the next impact arrives. Small increases in clearance, etc., will cause jamming or skipping that would not occur if there were more time between operating cycles.

Performance of Worn Intermittent Motion Mechanisms

Different types of intermittent motion mechanisms respond to wear in different ways. Some continue to function, but with errors in positional control,

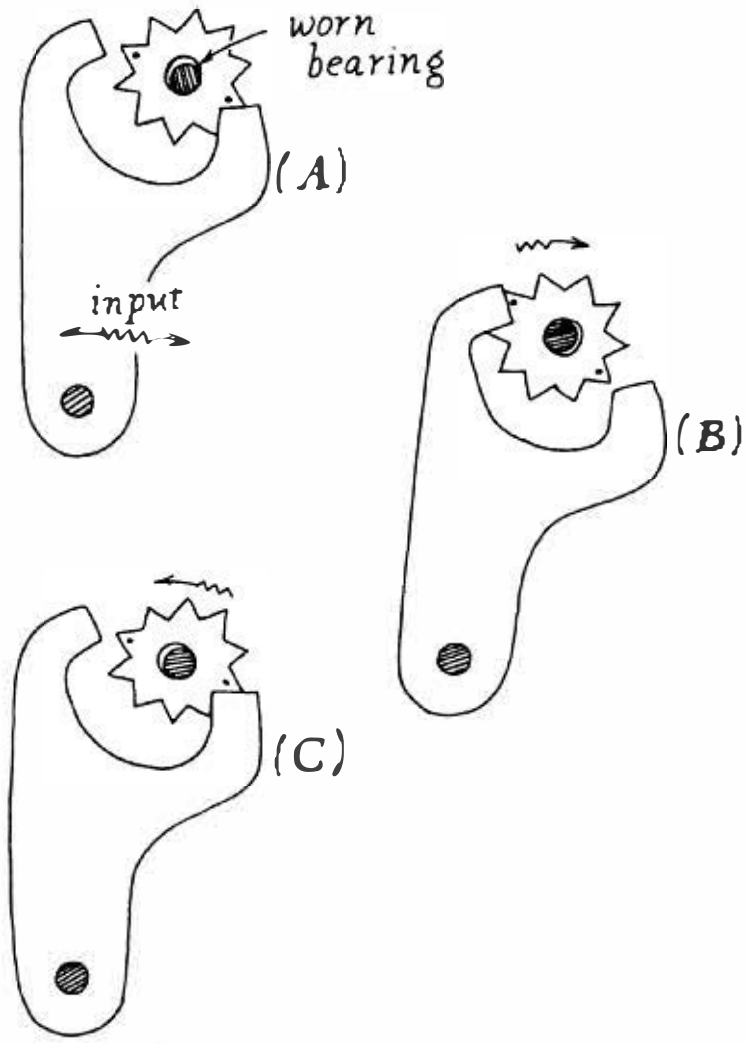


Fig. 5–7. Worn inverse escapement. The output wheel will "slump" (A) in its loose bearing between strokes of the driver. As a result, the wheel will often oscillate (B and C) instead of indexing. The wheel should index in a clockwise direction.

cyclic rate, or the like. Some continue to function but produce violent distortions in the system. Others just simply cease to function.

In Fig. 5–7, for example, an inverse escapement mechanism is seen, in which the bearing hole on the output wheel has been worn by repeated impacts of the driver against the wheel. On the first half of the stroke (part A), when the right-hand drive tooth comes in to move the wheel forward, it merely moves the wheel up from a slumped-down position due to the looseness in the bearing. Then, when the left-hand tooth engages the wheel (part B), it returns it to a slumped-down position. Part C is a repetition of the starting position (A) of the cycle. Thus the mechanism will no longer work—it just oscillates back and forth around a single drive tooth.

The ratchet of the system in Fig. 5–3, on the other hand, will continue to function even after it has worn badly; but the ratchet wheel will be indexed to different positions than it was when new. This may make the machine driven by the ratchet unusable, but since the ratchet is still capable of stepping the wheel, it may still have useful life in some situations.

The Geneva mechanism of Fig. 5-4 falls into a different category. Here the stroke is so long that it takes a great deal of wear in the drive pin, or slot, before the positional accuracy of the device is affected. But a little wear can drastically change the acceleration and velocity pattern of the mechanism. A "good" Geneva has a drive pin that slides in but fills the drive slot with no clearance or backlash. Such a Geneva exhibits the displacement-versus-time curve shown as a solid line in Fig. 5–8. A Geneva with a loose slot might exhibit the displacement curve shown as a dotted line in the same illustration: if it were driving a heavy, rigid, well-damped load.

If the system is partially elastic, however, and/or is operating at a high rate-of-speed, then it is possible for the worn Geneva to produce a displacement-time curve such as that shown in Fig. 5-9. As the load hesitates and jumps ahead of the driver, the pin contacts first one wall of the slot and then the other. So far, this does not seem to be too much of a problem. The mechanism is still functioning: the input pin is indexing the output slot with little positional error.

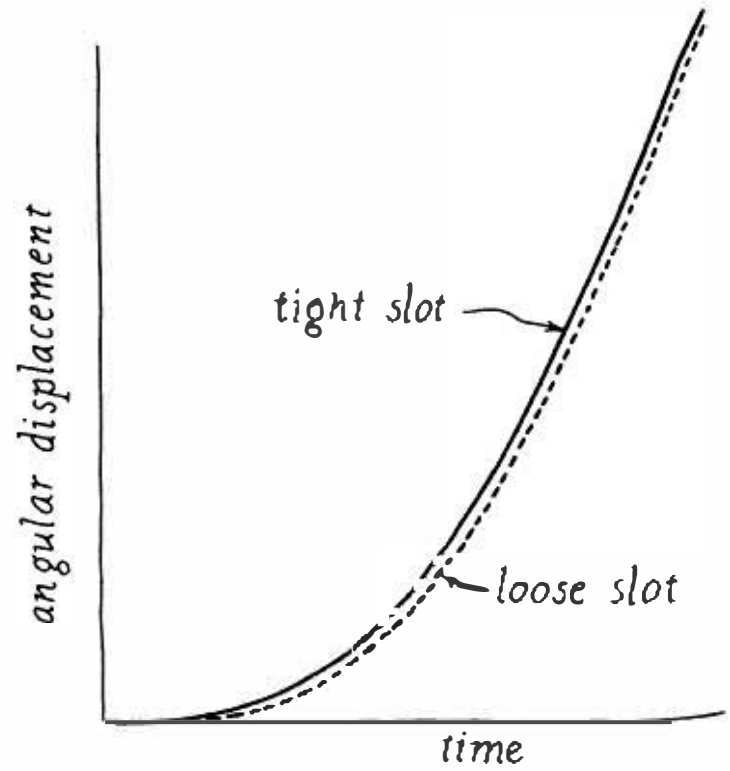


Fig. 5–8. Angular displacement-versus-time curve for a Geneva mechanism in which the roller is a snug running fit in the slot (heavy line); and for a Geneva mechanism with a loose roller-slot combination (dotted line); with a heavy, well-damped load and no elastic body reactions.

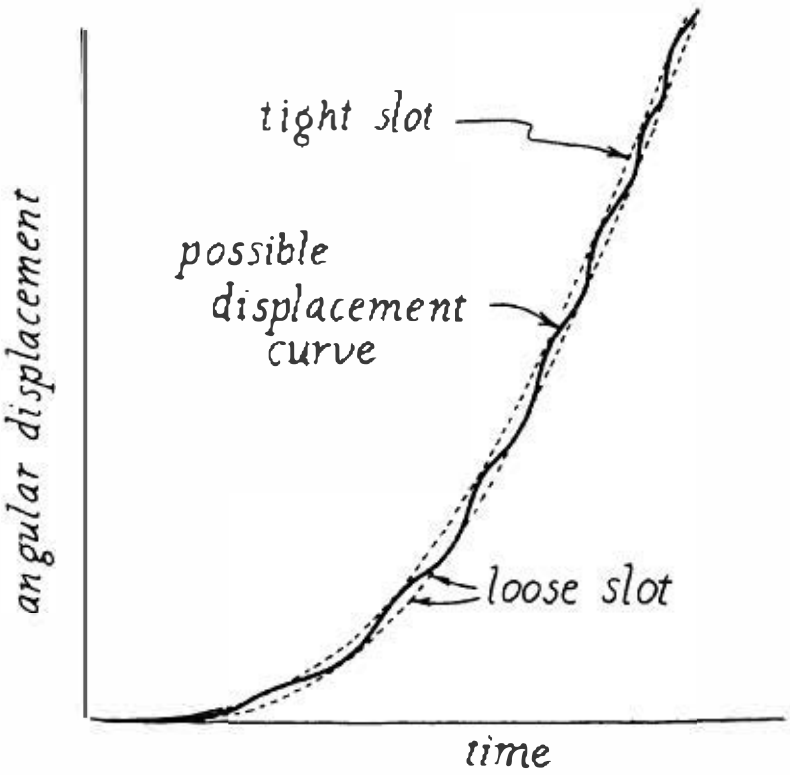


Fig. 5–9. Angular displacement-versus-time curve for a loose roller-slot Geneva driving an elastic load. Displacement of the output is shown as a heavy line. Compare with Fig. 5–8.

Graphical differentiation of this new displacement curve, however, reveals the problem, as shown in Fig. 5–10. The velocity and acceleration patterns are violently distorted from what they should be. (See Fig. 9–4 for comparison.) A little clearance between pin and slot has magnified and distorted the drive forces. New vibration stress and control problems will result. Wear rates throughout the machine will accelerate. Jamming, fatigue failures, or timing problems will finally result and the machine will cease to function.

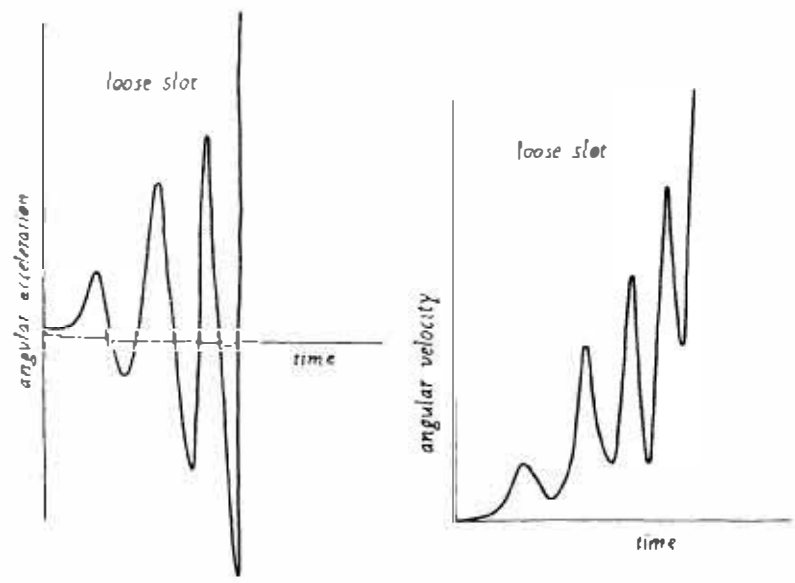


Fig. 5–10. Angular velocity- and acceleration-versus-time curves for the loose-slot Geneva driving an elastic load. These curves were obtained by successive differentiations of the heavy curve in Fig. 5–9.

Design Recommendations to Improve Stability

The importance of minimizing wear or the consequences of wear cannot be overstressed. Machines weighing tons are often reduced to scrap by wear-induced changes in geometry of only a few thousandths-of-an-inch on some critical part. There are several things that can be done to improve the life of intermittent motion machines. All of these things are intended to improve its stability in one or more ways; either to stabilize the geometry by reducing

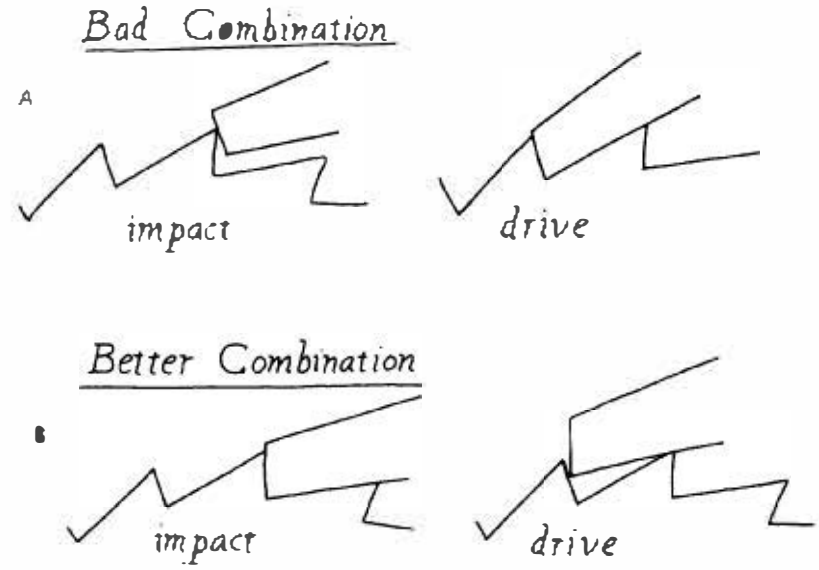


Fig. 5–11. Ratchet-tooth shapes to reduce wear. Ideally, ratchet teeth (and the teeth of other intermittent motion devices, such as gears, escapements, and clutches) should have large bearing surfaces at impact points to minimize impact stresses. It is also desirable to distribute drive forces over large areas, but since impact forces greatly exceed drive forces, large impact areas are more important than large drive areas.

wear, or to stabilize performance by making the design more tolerant of wear, or both.

The general "solution" to wear is the same as the "solution" to the problems considered in the last chapter. Keep stress levels low, minimize impact velocities, minimize sudden changes in velocity and acceleration, and the designs will work better and last longer.

More specifically, we can reduce relative velocities between sliding and/or impacting parts. Reducing velocities between sliding parts reduces adhesion and

fretting wear rate. This also reduces temperature generally speaking, and so reduces ploughing effects and the other types of wear. Reducing velocity under impact conditions, of course, reduces stress levels directly, as has already been seen.

Stress levels can also be reduced by increasing contact areas. This is especially important for areas of contact during collisions.

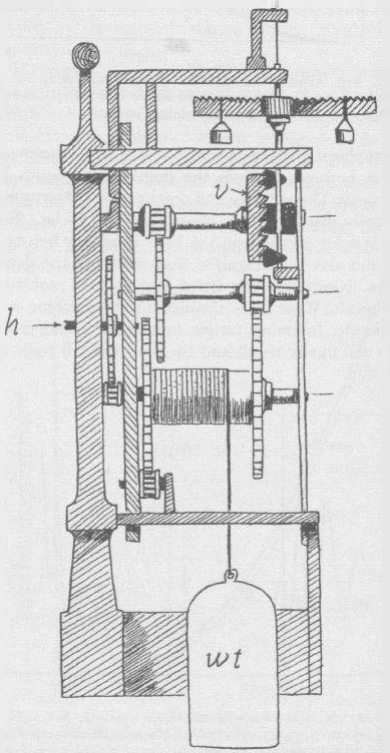
Ratchet teeth should be designed, for example, so that they will be loaded over flat surfaces when they are worn. This means that they subsequently run against sharp or rounded surfaces as shown in the middle view (B) of Fig. 5-7. It is not desirable to have ratchet or clutch teeth impact on sharp surfaces even if they push against flat surfaces as shown in the upper view (A) of the illustration. Ideally, of course, they should both impact and push on flat surfaces, as shown in the lower view (C), but if it is necessary to make a choice, the impact geometry should be favored since this is where the maximum stress levels are generated.

Another thing which should be done as often as possible, is to provide parts that are shaped to function properly even after they have begun to worn. In Fig. 5-7 there is an inverse escapement mechanism that had ceased to function after the bearing hole in the output wheel had been slightly enlarged by wear. It is possible to redesign this mechanism to reduce wear (a better bearing, for example), or to make the design more tolerant of wear. A center of rotation for the pallet arm could be selected, and tooth and wheel tooth geometries, that would reduce

The best way to study the wear tolerance of a new design is to make paper or plastic cutouts of the various links of the mechanism, showing the original dimensions of the parts at all wear limits. Put these pseudo parts through their paces on a drafting board and look for interference, jamming, loss of control, etc. This is similar to a tolerance study, of course, but with additional wear-induced dimensional changes accounted for. It is surprising how few designers study the effects of wear in this way, yet very often a design could be improved drastically by a few dimensional changes.

The designer should also consider supplying excess drive torque to accommodate the build-up of friction in a worn mechanism. He may have provided bearings that never wear, but wear points on "important" parts often end up where least desired and must be accounted for. Extra torque has extended the life of many mechanisms. On the other hand, extra torque can also limit the life of a mechanism, because extra input power means more power lost or unused somewhere else. If the load does not absorb it, the system will, as friction, impact, distortion, and wear. Thus the designer is caught in a dilemma. Maybe a little extra torque to provide for a little friction is

Of course the designer should also keep the service manual in mind. If wear cannot be eliminated, provide for maintenance of the critical elements of the machine. Provide adjustments or replacement of parts is common. Some heavy-duty Genevas, for example, are provided with replaceable "wear shoes" on each



side of each slot to prevent development of the motion patterns of Fig. 5–10.

Beyond this: test all designs thoroughly, recognizing and correcting original errors. If there is any company objection as to design time consumed, make it clear that you are now engaged in the "development" phase of research and development! Most managements assume that you (and all other "good" designers) will do everything right the first time and they may object to modifications. But intermittent motion devices are NOT predictable; testing and design development are almost always mandatory.

Historical Note

The amount of energy absorbed by an intermittent motion device determines such things as its efficiency and the rate at which it will wear. In general, of course, the more efficient the machine, the longer it will last and the better it will work. In this regard, it is interesting to see how far we have come in the design of clock and watch mechanisms.

The illustration (Fig. 5–12) shows a clock train built by Henri deVick between 1364 and 1370. The escapement is marked *v*; and *h* indicates the shaft on which the hands were mounted. These hands were about 3 feet long and probably weighed several pounds. The clock was intended for use in a municipal or church tower, as were all such clocks built at that time. The weight which drove this clock, labeled *wt* in the attached sketch, weighed 500 lbs and fell 32 feet in 24 hours. The input to the machine, therefore, was 16,000 ft-lbs every 24 hours! The output work, of course, was the rotation of the clock hands. They must have been cutting the gear teeth as they kept time! Comparing the efficiency of this clock to that of the modern electro-mechanical watch mechanisms that run for a year on a tiny battery shows how far we have come in our understanding of this class of intermittent motion devices.

CHAPTER 6

Experimental Techniques used to Evaluate Intermittent Motion

Exact Measurements Are Difficult

Since it is difficult, and sometimes impossible, to calculate exact forces, accelerations, velocities, and displacements in many intermittent motion devices, it is useful to know how to evaluate the performance of a prototype design, experimentally. Unfortunately, it is often as difficult to make exact measurements as to make exact calculations when we are dealing with transient forces of very short duration, deformations, and the like. Nevertheless, the combination of rough calculation and approximate measurement is still very helpful for evaluating and improving designs. The mere act of observing and measuring the performance of a model, of course, often gives the engineer insight that leads instinctively, if not scientifically, to an improved solution to his design problem.

In studying high-speed intermittent motion devices we need measuring equipment that can detect rapid changes in highly localized portions of the machine bodies being studied. Ideally, the measuring device should be stable (produce no vibrations, etc., of its own) and it should load a machine so lightly that it does not effect the motion of the thing being measured. Many of the techniques which satisfy these conditions require fairly expensive equipment, and some skill and experience to operate it correctly. We will not give detailed instructions for the use of such equipment here, since such instructions would soon be out-of-date and current instructions are available from the manufacturers of the equipment you purchase. We will describe the measuring equipment in general terms, however, so you will know what to ask for.

What Can Be Measured?

Displacement is probably the easiest parameter to measure at the present time. Many different techniques are available, experimental systems are relatively simple, and the results are generally easy to interpret.

Force is probably the next most easy thing to measure because such tools as strain gages and photoelastic coatings are available; but here, expertise is very definitely required. M.I.T. (and probably, other engineering schools) gives a two-week summer course in stress and strain measurement. This is an excellent way to get started if it can be arranged. Otherwise, work closely with your suppliers.

Velocity and acceleration are very difficult to measure directly, especially when dealing with transients, because there is very little instrumentation available. Furthermore, most of that which now is available really measures displacement or force, and then electronically integrates or differentiates these measurements to give a readout in terms of velocity or acceleration. Fortunately, we are usually more interested in displacement or in force, than we are

in velocity and acceleration, thus we do not usually suffer from the lack of velocity and acceleration measuring devices.

If direct measurements of force, acceleration, velocity, and displacement are impractical, or do not reveal enough information, there are other things that can be done. It is possible to measure the noise produced by a machine to evaluate the general quality of the device. This is especially useful for detecting incipient wear or abnormal clearances, etc. A model of the machine or of certain portions of the machine can also be made, and then measurements (force, velocity, displacement, etc.) can be performed on the model.

It is also possible to change the scale of the machine, while at the same time, slow down or speed up certain effects for more accurate evaluation. Scaling must be done by an expert in physics, however, if the final model is to be a true "analogue" of the original.

Last, but definitely not least, stroboscopic or photographic techniques can be used to help us observe the performance of the machine. One final point before proceeding with descriptions of experimental techniques. Some experimental tools are only good for measuring what might be called "milestone" events; that is, it could be determined how long it takes a mechanism to complete its stroke, (the end of the stroke being "the milestone"), or at what point in the cycle a clutch will actuate; the actuation of the clutch being the "milestone." Things that happen between the milestone events are not detected by the experiment.

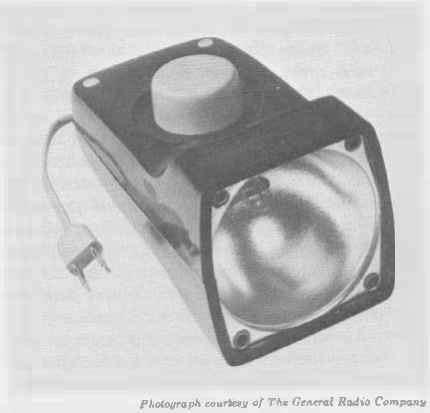
Other experimental techniques serve to "digitize" an event. We can determine where a linkage is once every five milliseconds, for example, but have no way of telling exactly where it is in-between these measurements. In some ways this is like milestone measurement but many of the readings will be made at relatively insignificant times during an operating cycle.

Finally, a few measuring tools produce a continuous analogue output telling us (within the limits of their accuracy) exactly what the machine is doing at all times. It is very important that the designer be aware of these differences, especially when dealing with intermittent motion devices where short-duration events and transients can be all-important.

Michelangelo is reported to have said th a all human understanding starts with a picture, eith er on paper or in th emind. Probably, no designer would dispute this. It is c y useful to get a good picture of your designs action. The two principal assistants here are th estroboscopic light source and

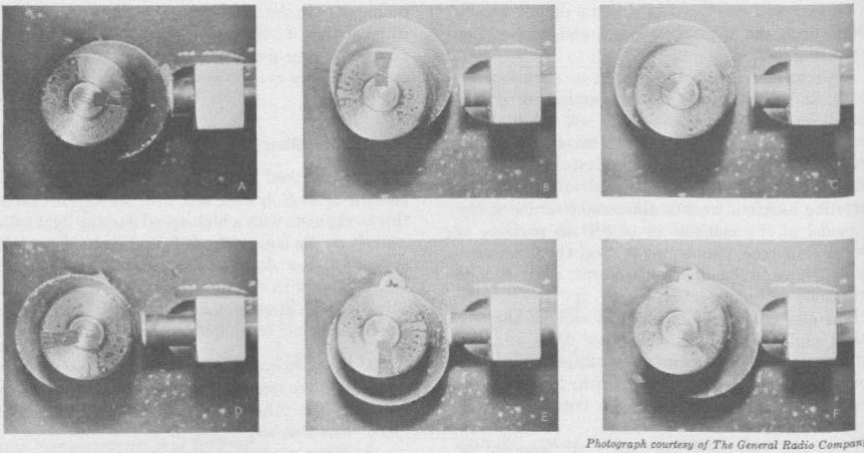
If a high-speed mechanism is repeatin ga certain pattern of mot over and over we can illuminate this mechanism with a high-speed flashing light called a stroboscopic lamp, Fig. 6–1, and make the motion appear to slow down or stop. Surely most readers are familiar with the technique. Following are some ints hen deciding whether or not to use stroboscopic techniques to evaluate your design

1. Stroboscopic lamps are relatively inexpensive and are very easy t ouse, compared to many of the other techniques t o be discussed in
2. When used alone, stroboscopic lamps are only useful wh en repet ve events. If used with one or more still cameras, however they are useful for evaluating single events. Whether used with or without a camera, the stroboscopic lamp is primarily a milestone measurement technique. By changing the point at which the light flashes occur, it is possible to detect how long it takes a ratchet



Photograph courtesy of The General Radio Company

Fig. 6–1. Stroboscopic lamp for visual or photographic observation of moving bodies.



Photograph courtesy of The General Radio Company

[illegible]

[illegible]uous readout [illegible] never possibl[illegible] wit[illegible]

4. [illegible]ugh [illegible] sometim[illegible] to believe, especially when the strobe rate is very high, the motion of the part being observed can appear to be continuous, thanks to the persistence of vision, but it is not. The experimenter is really [illegible] a collection of "still pictures" that often will not reveal short-duration reversals, deformations, hesitations, etc., unless these are repeated exactly, in each cycle, and are slow enough or gross enough to occur over a period represented by [illegible]

5. One very useful feature of the stroboscopic approach to life is that the experimenter obtains immediate feedback on the results of the experiment. If he cannot see the part being studied, he can move the light source, for example. This may seem like a trivial comment unless you have spent a lot of time and money on high-speed motion pictures, and have waited a week for your film to be returned from the developers only to discover that the camera was not in focus or that the part in which you were most interested was, unfortunately, partially concealed by shadows.

6. Stroboscopic lamps are also easy to synchronize with other timing tools or with the machine being studied, so that light flashes can be properly phased with respect to drive currents, clutch actuation pulses, limit switches, oscilloscope sweeps, etc.

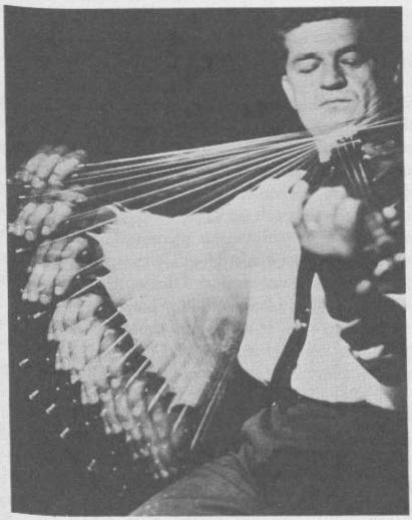
7. Stroboscopic lamps used alone produce no records to compare with future or prior experiments, but it is possible to take a series of still photographs of a repetitive event, using a strobe lamp as a light source and synchronizing the camera so that each photograph is taken at a slightly different time during the event (Fig. 6-2A through F). If the strobe is used instead of a mechanical photoflash, [illegible] be extremely short; down to something like a few microseconds, for example (compared to a few milliseconds for mechanical shutters). The result is a series of very sharp photographs that are excellent for making [illegible] a high-speed mechanism.

8. Single pictures taken [illegible] stroboscopic illumination, furthermore, are frequently sharp enough to capture such things as distortions and deformation of parts at the instant when impact occurs. This is one of the few experimental techniques that produces photographs sharp enough to permit accurate measure-

ments of such operations. With proper illumination, incidentally, several "pictures" of a moving object can be superimposed on one photograph, as shown in Fig 6-3.

9. Stroboscopic rates of up to 150,000 flashes per minute are possible, but the observer should remember that his eye cannot follow at that rate and he must use photographic techniques to capture short-duration events.

It is also very interesting to use closed-circuit television cameras in conjunction with stroboscopic illumination. Instead of observing the machine directly, the experimenter watches the TV screen. The stroboscopic flicker, which can be quite disturbing at low flash rates, is reduced by the persistence of the television screen. In addition, the picture tube will enhance or brighten the image revealed by the stroboscopic flash; in general, giving a much brighter, clearer picture than a high-speed motion picture. The large size of the TV screen also makes it easy for several observers to watch the experiment simultaneously. In fact, indeed, when there is more than one, this can be a very useful feature. Furthermore, video tape can be used to record these pictures for future observation, although, of course, they are recorded as magnetic signals rather than as pictures. Using video tape, we can even record single events stroboscopically and then, by playing them back over and over, turn them into repetitive events for careful study. Again, it is easy to synchronize the flash, the TV camera, and other devices such as oscilloscopes, for simultaneous observation of mechanisms and current or voltage traces. A typical set-up is shown in Fig. 6-4.



Photograph courtesy of The General Radio Company

Fig. 6–3. Single multi-position photograph taken with stroboscopic illumination.

And again, the experimenters should be warned that what he is viewing, although it appears to be continuous motion, is a series of still pictures. The TV system is capable of recording only about sixty frames per second, so once again short term distortions, reversals, or hesitations in the motion, etc., are usually overlooked. The image resolution, too, is lower than that of film so that accurate measurements are difficult.

Like other stroboscopic techniques, however, instant playback is possible (and with TV systems this includes instant playback of either a single frame or a continuous series of frames) so that the experimenter can tell whether he has successfully recorded the event he is trying to observe.

High-Speed Motion Pictures

High-speed motion pictures are another excellent tool for detailed observation of high-speed mechanisms. Like stroboscopic techniques they are best for milestone measurements, although by making measurements on each separate picture in a film strip, separately, one can obtain some "digitized" data.

Cameras can be used with anything from microscopic to telescopic lens systems to alter the apparent size of the device being studied, or to make observation from a distance possible. Typical cameras take pictures at rates varying from a couple of hundred pictures per second to tens of thousands of pictures per second. Films can be black and white, or color. Although this is difficult, cameras can be synchronized to machine cycles to make results more useful (Fig. 6–5). But high-speed motion pictures, must be used with caution, for just as strobe lamps, they appear to give a much better view of a mechanism than they actually do. One reason for this is that the camera only sees about 30 percent of the event it is photographing. The rest of the time the film is being moved into position to take the next frame. The camera, therefore, easily misses

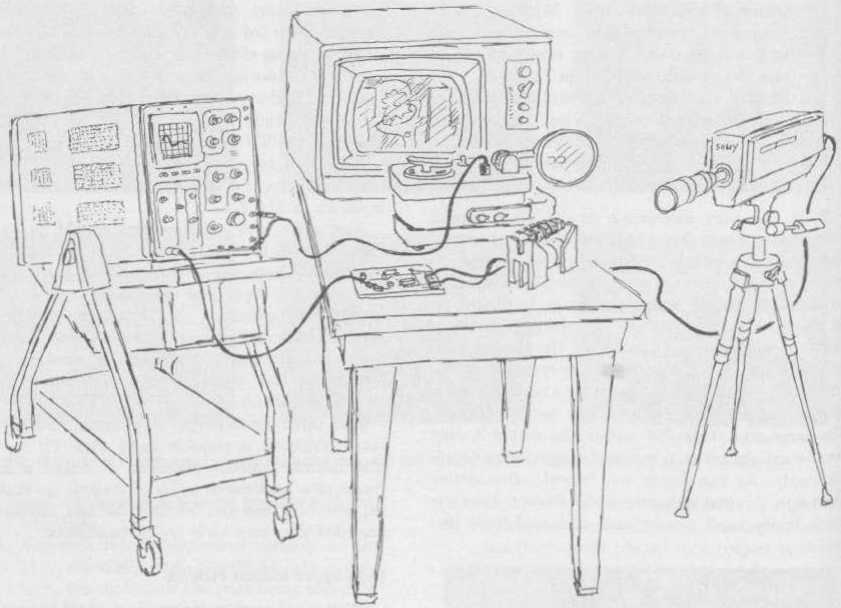


Fig. 6–4. Using closed-circuit television to aid observation of a stroboscopically illuminated mechanism.

small details of mechanism motion. Thanks to the persistence of the eye, the films appear to show a mechanism in continuous motion, but the experimenter should never forget that this is not actually the case.

Motion pictures are often blurred: the extent of blurring depending upon the reaction rate of the film and the actual velocity of the device being photographed. Slight deformations or distortions of parts, knowledge of which would be useful for evaluating impact phenomenon are often concealed as a result.

Photograph courtesy of Redlake Laboratories, Inc.

Fig. 6–5. New high-speed, 60,000,000 PPS camera for use in an industrial or scientific environment.

It is also difficult to synchronize motion pictures with other experimental printouts, for example, oscilloscope traces or strip-chart recorders. Cameras, in general, are not designed for synchronization in the way that stroboscopic lamps are.

Lighting, also, is a difficult problem. Many high-speed motion pictures appear to be quite gloomy because there is no way to provide optical enhancement of the image. Nor is there an immediate feedback on results to aid the experimenter. Unless you have your own facilities for developing high-speed motion pictures, it usually takes at least several days to get the films developed and to arrange to view them. If you have made a mistake in your lighting, or in synchronizing your camera with the events to be photographed, you will have to start over and will have lost several days in the process.

Nevertheless, high-speed motion pictures remain [illegible] good method [illegible] the performance of [illegible] devices. They can be used to photograph single or repeated events and you can analyze the resultant pictures, frame-by-frame if you wish.

[illegible]

Frequently, the experimenter will want to know [illegible] the displacement [illegible] function [illegible] formation, of course, he can graphically [illegible] imate what the velocities, accelerations, and forces must be. In many cases, however, he may only care about displacement. [illegible] it is supposed to do in some [illegible] Do the various parts of the machine [illegible] n under control? Is the machine subject to [illegible], hesitation, vibration, etc., which were not planned by the observer?

Measurements from High-Speed Movies

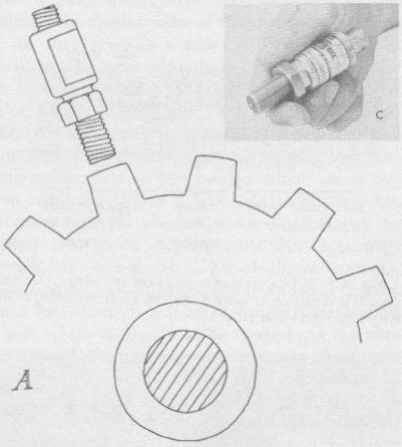
One way to get the displacement of a part as a function of time is to make measurements of this motion as recorded by high-speed motion pictures, frame-by-frame. In order to do this we must paste or paint targets onto the moving device and onto some part which remains grounded or fixed. Measuring the relative displacement between these two [illegible] and knowing the speed of the camera, a rough measurement of displacement as a function of time can be estimated for the part in question. It is usually wise to photograph some sort of timing device at the same time you are photographing the object, however, a smost high-speed cameras start relatively slowly, accelerate to a maximum speed, [illegible] rate, therefore, varies during the film strip and it is [illegible] to count frames to determine time, if small errors would have a large effect on your resulting decisions.

[illegible]

Figure 6-6A shows another way of measuring displacement as a function of time. A magnetic pickup (C) detects (without contact) the tops of gear teeth as they go by on a rotating shaft. A strip-chart recorder indicates a voltage pulse, as in B, for each tooth that passes the cell. By counting pulses, the experimenter can determine the rotary displacement of the gear as a function of time. This is a [illegible] technique and has a lot of [illegible] precedence. The first attempt at a scientific measurement must have been to count something.

If two pickups are used, then reversals of motion can also be detected as phase [illegible]s in the output pulse train by using a dual-track [illegible], as shown in Fig. 6-7.

Counting gear teeth is acceptable only if we are looking for crude measurements of motion, or are trying to determine how long it takes to reach certain [illegible]. If a more detailed picture of an event is wanted we have to resort to a better transducer. Figure 6-8 shows a linear encoder, a piece of glass on which have been printed thousands of black lines. A photocell reads these lines to measure linear displacement. The technique is basically the same as gear tooth counting. Rotary shaft encoders employing the same principle are also available; some



Photograph courtesy of Tempo Instrument Division, [illegible] Equipment Company

[illegible] pick-up [illegible] motion by counting gear teeth. [illegible] pulses can be accumulated on a strip-chart [illegible], B) above. The photograph (C) illustrates a typical pick-up

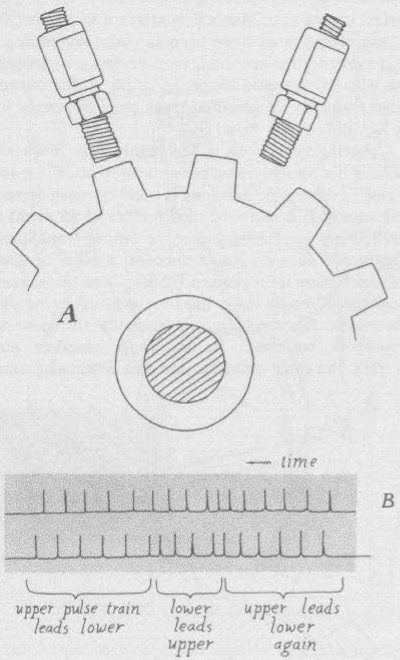
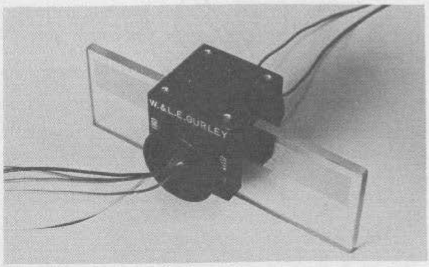


Fig. 6–7. If two proximity pick-ups are used (A), then reversals in direction of motion can also be determined, as in the strip chart shown in (B).



Photograph courtesy of the W&LE Gurley Company

Fig. 6–8. Linear optical encoder. Optical encoders are similar in broad concept to the gear-teeth counters of Figs. 6–6 and 6–7, but can produce many hundreds or thousands of

of them will produce as many as two million pulses per revolution of the input shaft.

Figure 6–9 shows another type of shaft encoder. This one uses what are known as Moiré patterns, to reveal very small displacements. Two glass discs, (as in A), marked with fine line patterns, are superimposed on each other. One disc is fixed to "ground." The other is mounted on the rotating shaft whose motion is to be studied, as shown in C. One disc has one more line than the other, and an

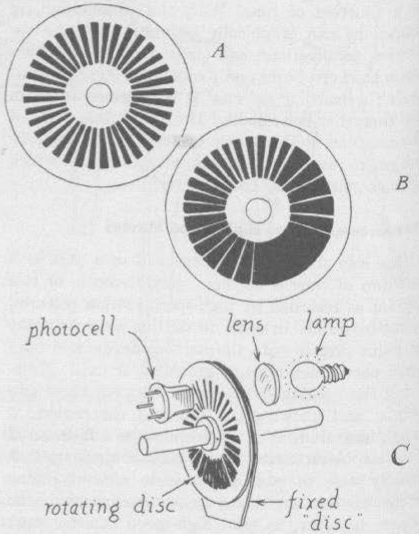
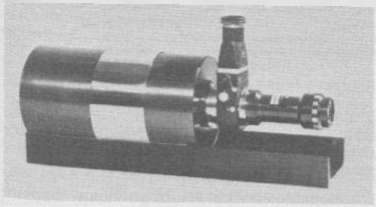


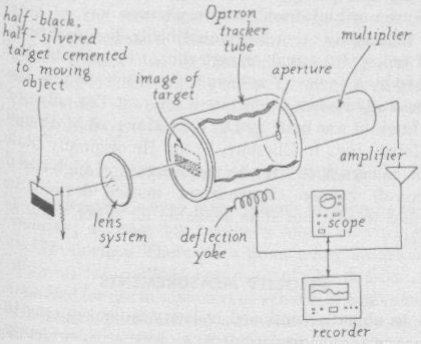
Fig. 6–9. Another optical encoder. This one uses two discs and develops what is known as a Moiré pattern that produces large optical changes for very small displacements.

interference pattern B, is created. Rotation of the moving disc through an angle representing the thickness of one line results in one full revolution of the dark interference pattern. Since the interference pattern is quite broad (nearly 180 degrees in extent), it is easy for a large photocell to read very fine motions of such a disc. Again, two cells could be used to detect phase changes, and hence, reversals in motion. For linear motion, experimenters often use optical gratings to produce Moiré patterns. So-called replica gratings (photographs of true gratings) having up to 30,000 lines per inch are available for a

[illegible]

Another method of measuring fine displacements is to cement a piece of magnetic tape onto the moving part, having prerecorded the tape with several hundred, or more, magnetic [illegible] on a digital tape recorder. Displacements of the part are now measured by reading these [illegible] with a conventional tape-recorder head. If two magnetic tracks are used, phase changes aga[illegible] moving [illegible] produce a signal which is proportional to the rate-of-change of magnetic flux [illegible] and with slow motions little [illegible]-change. Conventional magnetic heads, therefore, will not produce signals during dwells, and the like. A flux-responsive, or Hall effect magnetic head, [illegible] motion [illegible]





Drawing based on information received from the Optron Corporation

[illegible] 6–10. Optical [illegible]. A half-black target is cemented to the part whose motion is to be recorded. This system produces an output voltage which is proportional to the displacement of the part.

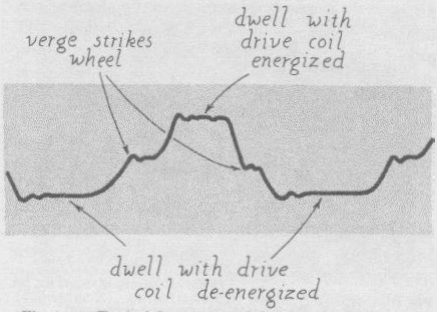


Fig. 6–11. Typical Optron record showing the displacement of the verge arm of an inverse escapement (See Fig. 12–22) as a function of time.

Optical Tracking

The devices which have been discussed so far "digitize" the displacement of the part being studied that is, they produce a series of electrical pulses, each of which represents a finite displacement of the part. In this sense, they are rather like the high-speed motion picture and stroboscopic techniques discussed earlier. We are getting a series of still pictures; and if our measurements are too crude, they can miss subtle refinements in the motion which occur between readings. If this is unsatisfactory, there are instruments on the market which can be used to measure continuous displacement. The apparatus shown in the photograph in Fig. 6–10, for example, is called an Optron. It projects a small beam of light onto a half-silvered, half-black target cemented on the part whose motion is to be studied. As the part moves, the Optron automatically tracks the target and produces a strip-chart record of the displacement of the part as a function of time. It will only track motion along one axis, as shown in the diagram (and, therefore, must be used on reciprocating links rather than rotating parts), and is sensitive to motions between 12 microinches and one inch. Nevertheless, it is a good way to measure either repetitive or single events. Figure 6–11 shows a copy of a typical Optron-produced record of the motion of the verge arm of a small inverse escapement.

The Photo-Tech apparatus shown in Fig. 6–12, is similar to the Optron, but can measure motions in more than one axis, multaxially. Again, the output is a continuous trace rather than a digitized trace. Also, both optical tracking devices can see reversals and dwells as well as continuous motion.

[illegible]

[illegible]

There is one type of photograph which can be used to give a continuous (rather than digitized) record of a high-speed event; the so-called drum camera shown schematically in Fig. 6–13. A strip of film is fastened to a drum which is rotating at high speed. A dowel pin, or other highly reflective cylindrical target, is cemented to the part whose motion is to be studied. The part is then illuminated by a single high-intensity lamp.

Reflections from the dowel pin are focused by a lens system and, passing through a cylindrical lens are imaged as a spot of light on the film fastened to the drum. As the part oscillates and the drum turns the image moves back and forth on the film. A fixed dowel-pin target must also be provided on the ma-

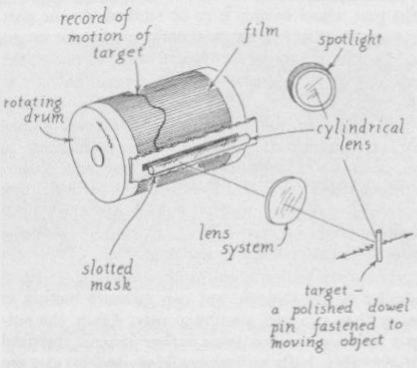


Fig. 6–13. Schematic of a high-speed drum camera.

[illegible], produces a track on the film. The displacement between these two marks on the film is a direct measure of the displacement of the part. A shutter (not shown in the illustration) exposes the film for one revolution of the drum. One or more cycles of motion, depending upon the relative speed of drum and object. Several parts on the same machine can be photographed simultaneously if they are close enough together. Rotating parts also can be studied, as well as reciprocating parts, by fastening them with several dowel pins which appear one after another on the film.

This technique produces a film strip such as that shown in Fig. 6–14. Because it is continuous, it can reveal even very short-duration events; sudden stops, reversals, vibrations, etc., in the parts being studied. In fact, I have been amazed at how much more was revealed by a streak camera than by even such

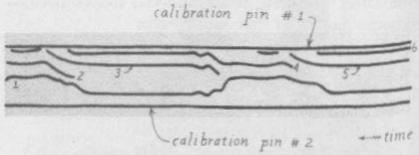


Fig. 6–14. Film strip made on a drum camera showing the motion of six target pins and two stationary calibration pins. Pins 2 through 6 are spokes in an indexing wheel, and thus, disappear from view.

sophisticated optical tracking equipment as the Optron or high-speed motion pictures.

Like many other techniques being discussed here, of course, the streak camera cannot be assembled or used by a novice. You have to find someone who owns one and knows how to operate it for you. The only one I know of was built by Dr. Karl Mann, of Math and [illegible], Conn. He originally built his camera to study the performance of high-speed aircraft engines (intermittent motion devices, in other words) but it is available for other work as well.

[illegible]

As already mentioned, velocity is far less easy to measure than displacement, at least with presently available transducers. One common technique is to use a tachometer, which is an electrical generator whose output voltage is proportional to the speed at which it is driven. This is a very acceptable device

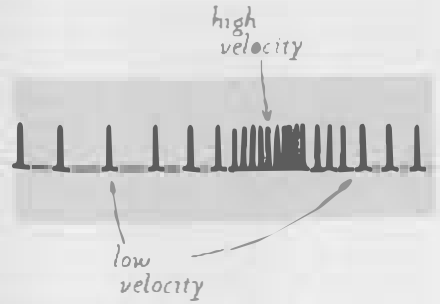


Fig. 6–15. Strip-chart record used to determine velocity of a rotating part. The higher the density, or frequency of pulses, obtained from a displacement transducer, the higher the velocity of the part.

to use for a steady-state machine, but not very useful for intermittent motion where we have to contend with starts, stops, reversals, and dwells, as well as with rapid motion. Most tachometers are very poor measurements at low velocities, for one thing, and most do not distinguish between clockwise (CW) and counterclockwise (CCW) motion.

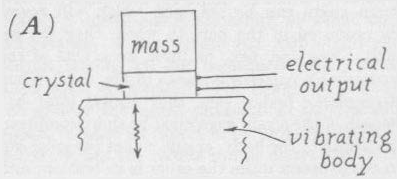
We can use some of the displacement transducers we have already discussed, however, and electrically interpret the results as velocity rather than displacement. For example, with the tooth-counting techniques (and more sophisticated but similar digitizing techniques involving Moiré patterns, magnetic tapes, etc.) we can measure the frequency of pulses generated by the transducers to determine the velocity of the part, before then counting the number of pulses to determine displacement. In effect, what we are doing is electronically differentiating the original pulse train. Perhaps the technique can be better understood by reference to Fig. 6–15.

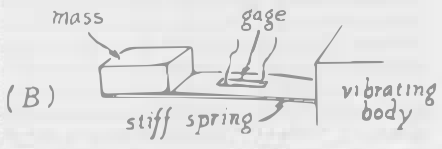
ACCELERATION MEASUREMENTS

Acceleration is even more difficult to measure than velocity; at least for relatively small intermittent motion devices. There are a great many accelerometers on the market, but most of the devices are intended for use in some sort of vehicle, or equivalent, where acceleration changes are relatively slow. Such things as the spring-mass, transit time, bubble, gyro, or vibrating spring accelerometer are good for vehicle motions but are rarely used by machine designers.

One accelerometer which is useful on machines, however, is the mass and piezoelectric crystal device. A schematic of one of these is shown in Fig. 6–16A. The greater the acceleration the more the mass presses on the crystal and the higher the voltage produced by the crystal. A mass, stiff spring, and strain gage accelerometer might also be used, as shown in Fig. 6–16B. Both of these accelerometers are really measuring force, or torque, not acceleration; aren't they? We have merely built a simple [illegible] in which acceleration results in a calibratable force, and hence, is easy to measure.

We must be careful that the stiffness of the instrument is great enough so that it will not vibrate during the experiment, because, as has been seen (Chapter 3), vibrations can amplify forces and give false readings. It is best to have a very stiff spring in the measuring instrument so that the equipment will operate on the low end of the vibration curve (Fig. 3–14) and the response of the instrument will be on a one-to-one basis with the response of the input. Making the spring stiffer, however, reduces its deflection and reducing its deflection reduces the output of the transducer; whether we're using a piezoelectric crystal, or a strain gage accelerometer. This sometimes forces us to use a great deal of electronic amplification in our measuring system. Unless the amplifier is a very good one, stable with temperature and insensitive to changes in power supply levels, etc., it will distort the results obtained





[illegible] tion of machine parts. The piezoelectric accelerometer (A) can be obtained in very small sizes and thus is useful for measuring the accelerations in instruments as well as in larger machines. (B) Mass, stiff spring, and strain gage.

from the original transducer. The whole [illegible] rat ber complicated and great care should be taken [illegible] the experiments and [illegible]etung the [illegible]

[illegible]

Force measurements, just as acceleration measurements, should be made by well-trained people, since these measurements involve such things as strain gages, photoelastic materials, or piezoelectric effects. This does not mean that the "average experimenter" cannot learn to use such techniques, it does mean that he cannot use them without some prior education obtained from a textbook, a classroom, or the manufacturer of the instrument used in the experiment.

[illegible]

Forces can be measured if they are high enough to deform something [illegible] force measuring devices really respond to deformation, not to force directly. A strain gage, for example, is a fine piece of wire whose resistance is proportional to its shape. As the wire is stretched, its resistance increases and this change can be interpreted in terms of force applied on the part to which the gage is cemented, if we know the stress-strain relationship for the part itself.

Strain gages can be obtained which will reveal stress produced in the part to which they are cemented, along one, two, or three axes. One of the many possible patterns is shown in Fig 6-17. When used on elastic bodies (and they always must be the length of the strain gage must be short compared to the length of the body, or one end of the gage can be in compression while the other is in tension, and the results can be very misleading, especially when the body is subjected to impact (see Chapter 3). If small gages are used on large linkages, however, results can be quite informative. Again, training and expertise are required to interpret results properly.

The electrical signal produced by a strain gage is continuous, and can be recorded on a [illegible] strip-chart recorder or oscilloscope. This gives it a big advantage over photoelastic techniques (discussed next) in many situations, because even though the photoelastic material changes its pattern in a continuous fashion, the only means we have for recording these changes are discontinuous (digitized) and important portions of the event can be missed.

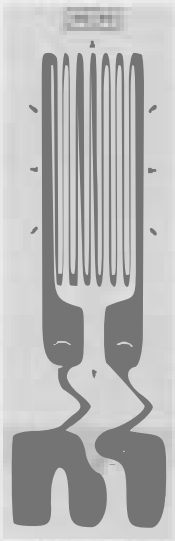


Fig. 6-17 Typical pattern of a [illegible] strain gage.

Photoelastic Coatings

Another method of detecting the forces on a machine body is to coat the part with photoelastic materials, and by observing colored patterns in these materials determine the deformations, and hence, the stresses to which the various portions of the body are subjected. Originally, people built plastic models of the parts to be studied and then studied the photoelastic patterns in the models in an attempt to determine to what stresses the actual machine parts must be subjected. It took a great deal of expertise to relate the performance of the plastic models to the performance or quality of the actual machine. This is still a popular technique, but fortunately, it is now possible also to coat the machine, itself, with photoelastic materials, and thereby to reveal the stresses to which the actual part is subjected while in use. Again, expertise is required to interpret the results, but one difficult step in the process has been eliminated.

This technique has the great advantage, in an elastic body situation, of revealing the local stresses at all points over the surface of a body and not the stress at just one particular point. In impact situations we can get stresses traveling through the body in surge waves, reflecting from free surfaces, interfering with and reinforcing stresses produced by subsequent impacts, etc., so the photoelastic technique proves, indeed, to be a very useful one. Incidentally, photoelastic coatings reveal local stresses to a microscopic level, and (if used by someone who knows what he is doing) can show the difference between the stress levels in the tip and root of a fine ratchet tooth, etc.

The photoelastic material also responds instantaneously to stress levels. The fastest high-speed cameras are unable to detect any lag between applied stress and the appearance of a pattern in the material.

Here then, is a material which appears to be ideal for our purposes, the evaluation of short-duration localized effects. Unfortunately, however, we have to fall back on high-speed motion pictures or stroboscopes, etc., to see or record rapidly changing photoelastic patterns. And this gets us back into difficulty, because, once again, we are looking at a series of still pictures that appear to be, but are definitely not, continuous. Again, rapid changes that would have extreme significance for us, can be completely overlooked. But it is clear that photoelastic techniques are still extremely useful.

Torsion Bars

Initially it was stated that force levels can be measured only if they are high enough to deform something. In many cases, forces are not high enough to deform the parts of the actual machine studied. In such cases, either something else, such as displacement, must be measured and a graphical estimate made of what the forces must be, or sometimes a deformable link in our mechanism train can be provided for the sole purpose of revealing forces that are transmitted through the train. One example, shown in Fig. 6–18, is a so-called torsion bar which is introduced between a driver and load to reveal torque. The bar twists as the driver attempts to move the load. The higher the load the greater the angle of twist. A sensitive displacement transducer of some sort detects this deformation and produces an output signal which can be interpreted in terms of torque if we have calibrated the torsion bar and transducer properly before the experiment. The

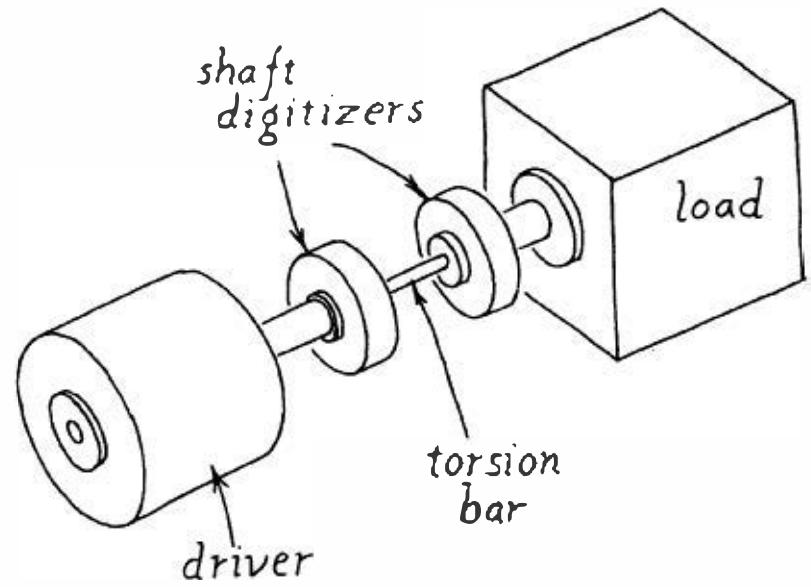


Fig. 6–18. Introducing a torsion bar between driver and load to measure torque.

transducer, in this case, can be anything that detects very slight displacements or deformations; including high-resolution shaft encoders, strain gages, photoelastic coatings, etc.

As was the case with the acceleration transducers discussed earlier, the stiffness of the torsion bar must be high enough to prevent it from introducing spurious oscillations in the system. Again, we want to work at the left end of the vibration curve, as shown in Fig. 3–14, where there is nearly a one-to-one relationship between the motion of the mass (in this case the load) and the input. Of course, if there were exactly a one-to-one relationship, we would not get deformation of the torsion bar and would get no output reading on our transducer, thus, we must allow some deformation. The more that is allowed, the greater the signal produced in the transducer. Again, it takes an expert to perform this experiment correctly, but the results can be extremely valuable.

Servo Torsion Bar

It is also possible to build an electrical torsion bar using a torque transmitter and torque receiver, as shown in Fig. 6–19. This is a servo-mechanism drive system. The average voltage across the electrical lines connecting transmitter to receiver is a measure of the torque being transmitted through the system. Again, the system "stiffness" must be high or oscillations (vibrations) which will distort the results will be introduced. It helps, therefore, to drive the transmitter with a slow-, and very constant-speed motor and keep the operating voltages high enough to reduce the lag between receiver and transmitter.

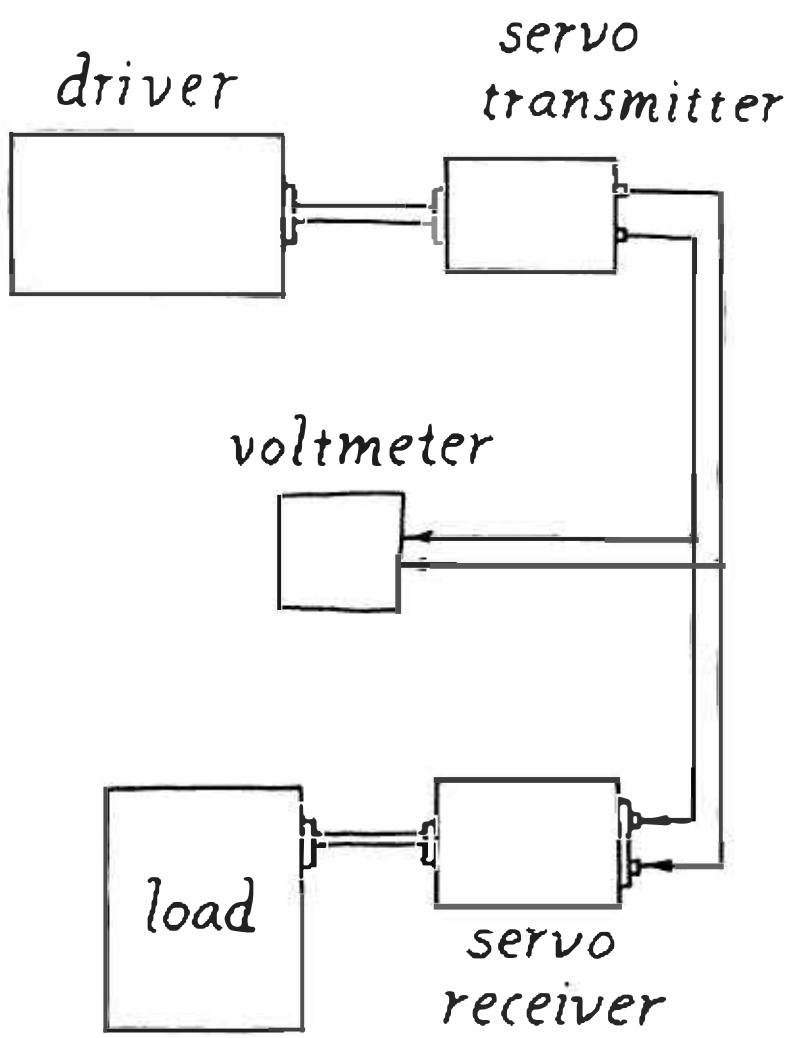


Fig. 6–19. Servo mechanism "torsion bar." This is the electrical equivalent of the system shown in Fig. 6-18.

An electrical torsion bar of this sort is very useful when it is desired to measure torque in some remote location such as an oven, or deep inside a machine.

NOISE MEASUREMENTS

Another method of getting a general feeling for the performance of a machine is to measure and evaluate the noise it makes. Worn or broken parts will produce different, and usually louder, noises than close-fitting or correct parts.

There are several types of noise-measuring devices. Some only measure the general amplitude of noise and are not very useful for evaluating machinery. Others will isolate and measure the amplitude of different noise frequencies contained in the total noise made by a machine. These so-called spectrum analyzers can be useful. I think the most useful of these is the so-called voice-print equipment which we are told will someday replace the fingerprint as a means of identifying criminals. Of course you cannot tell by looking at a picture of a noise whether or not a machine is functioning properly, because not being used to seeing this kind of a record, you have no experience for relating it to a machine performance. Noise analysis of a machine is only useful after a great deal of noise records have been accumulated and studied thoroughly, and you have simultaneously used other, more conventional, means of evaluating machine performance, all the while relating conventional results to noise-print results in order to understand just what the noise print is trying to tell you. Several years of experience are usually required before the noise print can be used directly on a new machine to determine, for example, whether or not it might fail sooner than its fellows. But since the noise print simultaneously listens to all portions of the machine (although you can direct its attention to one portion of a machine or another, if you wish), since it is always a non-contact and nondestructive measuring tool, since it can listen to things that happen inside the machine as well as on the surface, etc., it can be an extremely useful and powerful tool if the time is taken to learn how to use it.

Multiple Experiments

It is frequently helpful to use several different transducers simultaneously. Several different displacement transducers used in different parts of the machine, for example, can reveal phase lags, vibrations, hesitations in one part that are not reflected into another, etc. Simultaneous measurements of displacement and velocity and/or force can also be revealing. It is also useful to record things such as drive currents, clutch actuation pulses, etc., in the machine, while displacement, force, or velocity measurements are being made so that a machine cycle can be related more accurately to your transducer records.

Models

Several different types of models can be used to help us evaluate machine designs. The oldest and simplest is a hand-operated physical model of a portion of the machine. Today such models are usually made of clear plastic and at an enlarged scale to make relationships between parts easier to see. After putting the model through its paces in a "to print" form, try opening bearing holes, or shaving a little material off possible wear points to see how wear will alter geometric performance. The foot-high

model in Fig. 6–20 (of a mechanism that is only an inch long in the actual product) demonstrated very clearly that 0.005-inch wear in the wheel bearing—0.080 inch in the plastic model—would allow the wheel to "slump" enough between strokes of the drive verge to prevent motion. This design was discussed and illustrated in Chapter 5. The verge keeps entering the same teeth, over and over. The problem once revealed by the model, was solved by [illegible] the wheel bearing, greatly extending the life of the entire machine.

Another useful type of model is a mathematical model in the form of a computer program. As we have seen in previous chapters, this is difficult to build since such things as coefficients of restitution or friction cannot really be predicted. But with a digital or analogue computer we can sometimes use a mathematical model in conjunction with a physical model to determine what these coefficients, or other unknowns, must be. The procedure is very simple: Write a computer program for the problem, leaving [illegible] to be expressed as a single letter added or multiplied by one, because you do not know "exactly" what the term should be. Write a subroutine program for these equations, with provisions for manipulating the unknown terms.

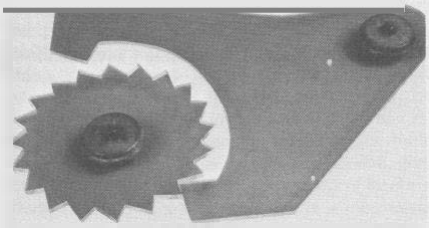
Now test the performance of an actual model of this same device, using one or more of the experimental techniques described in this chapter. For example, you might obtain an Optron record of the displacement of the links as a function of time. Measure the [illegible] for the values of the unknown terms or parameters. Compare the resulting computer output (a prediction of motion-versus-time, for example) to the actual performance of the sample machine. The two will probably differ by quite a bit. Now manipulate the data inputs to the computer program; try a larger coefficient of friction, for example, and see what this does to the performance predicted by the computer. Is the agreement between the experimental and actual performance curves better or worse? The answer to this question will tell you what to try next.

This technique is very useful in improving mathematical models. Often, you will find that it is impossible to make the computer results fit the actual results. This usually means that your basic equations are wrong; you do not just have incorrect values for some of the unknowns. By determining what the equations needs to make it behave, you often gain insight into unsuspected factors in your design.

Of course we are not interested in simply improving a mathematical model. But once we have achieved an agreement with the [illegible] model, we can now turn around and use the mathematical model to make significant improvements in the design—with understanding.

Another useful type of model is a physical analogue of the design. This is some device or system whose performance can be defined by mathematical equations that are identical in form to the equations which describe the machine being studied. For example, the performance of a mass-spring-dashpot system can be duplicated by the performance of an electrical circuit involving inductance (an analogue of mass), capacitance (the related analogue of a spring), and resistance (the related analogue of viscous friction). We can use an analogue computer or a digital computer program that simulates an analogue computer or we can assemble our own electronics circuit. This technique, like most of those discussed in this chapter, requires skill and training but can be very useful.

One unusual type of analogue model is shown in Fig. 6–21. This is a large model, nearly 18 inches high, of a watch escapement that is actually only three-quarters-of-an-inch long. The large model in this case is not just an enlargement of the small design for geometric studies, rather it is a true kinematic analogue of the small mechanism. The balance wheel works, in slow motion, and its momentum transfers, velocities, etc., are all interrelated in the

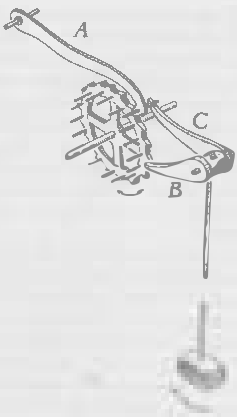


[illegible] of this sort are useful for studying geometric and dynamic problems.

same way these parameters are related to each other in the small model. It is the best presentation imaginable. But it is not just a blow-up of the real design. Thicknesses, materials, and spring constants etc., are altered to make the a true mechanical analogue of the original design. And it took a good theoretical engineer a good part of a year using dimensional analysis, to design the analogue so that this would be true. Mechanical analogues of

Photograph courtesy of The Raymond Engineering Company

ment. This is a kinematic (rather than just geometric) analogue of the actual design.



mechanical designs are common in aeronautics (wind tunnel models) but most machine designers are not aware of this technique as it relates to other fields.

Figure 6–22 is a sketch of a clock escapement that bears some resemblance to those in use today, but it was designed by Galileo. His goal—better measurements: more accurate experiments. His means (and the means of every experimenter until the late 19th century)—intermittent motion mechanisms: mechanical escapements.

So much for experimental methods. I have not attempted to be "complete." There are many, many experimental techniques for measuring force, acceleration, velocity, and displacement that I have not even mentioned. But there I have mentioned. I think, are particularly suited to the odd world of intermittent motion

CHAPTER 7

Ratchets

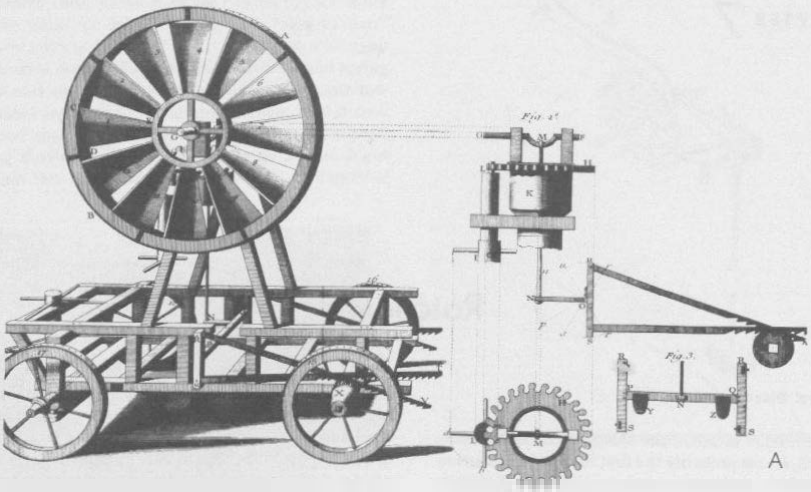
General Discussion

The ratchet is one of the oldest of all mechanisms. As such, it was probably the first intermittent motion mechanism, and was certainly the most common during the first century of the industrial revolution. Leonardo Da Vinci's notebooks are full of ratchet applications, and they were probably in use centuries before he came along. Most of these early applications, however, seem to have used the ratchet for mechanical advantage rather than to produce intermittent motion, the latter really being a product or need of the industrial revolution. Leonardo's ratchets, for example, are usually used to control engines of war; a man winds a catapult or crossbow, and the ratchet allows him to do this in short, easy steps, resting (dwell!) between exertions (index!). The drawing of the early machine (Fig. 7–1A), shows ratchets used to convert the continuous, relatively high-speed rotation of windmill sails into the slower, nearly continuous rotation of the rear wheels of a vehicle. I am sure the designer (who was not Leonardo, in this case) would prefer both input and output to be continuous here, but he settled for the wind and the ratchet. Figures 7–1B and 7–2A and B, showing some of the demonstration models built about 1876 by The Science Museum in London, give some indication of the number and variety of ratchets which were known a long time ago. It is interesting to note that such things as Genevas, star wheels, and escapements are also included in these photographs of "ratchet trains."

In any event, the ratchet is a logical place to start a discussion of different types of intermittent motion mechanisms. Another thing that makes it a logical candidate for first place in this discussion is its simplicity. A toothed wheel, a pawl, and a lever are all that is required to make a simple ratchet, as shown in Figs. 7–3 and 7–4. The first of these illustrations, Fig. 7–3, also shows that a ratchet can be built with (A) compression pawls, or (B) tension pawls. Figure 7–4 shows that the input can be through the pawl or through the wheel; and that internal as well as external ratchet wheels are possible.

Simplicity is one of the big advantages of the ratchet. Other related advantages include low cost and reliability. The ratchet is also noted for its ability to carry a large load in relation to its size. It is also a versatile device and is used in an amazing number of applications ranging from moderately heavy-duty machinery to high-speed instruments.

Disadvantages of the ratchet include the fact that it is an impacting mechanism, as seen in Chapter 5. There are ways to reduce the impacts in certain versions, but impact will almost always be present to some extent, and can lead to wear, control, and stability problems unless the rest of the system is properly designed. The basic problem, of course, is that impacts produce forces throughout the mechanism that are well in excess of the subsequent drive forces, as shown in Fig. 7–5, A and B. Impact also results in noise, which is very undesirable in most applications, and "noise pollution" should soon get a lot more attention from machine and instrument



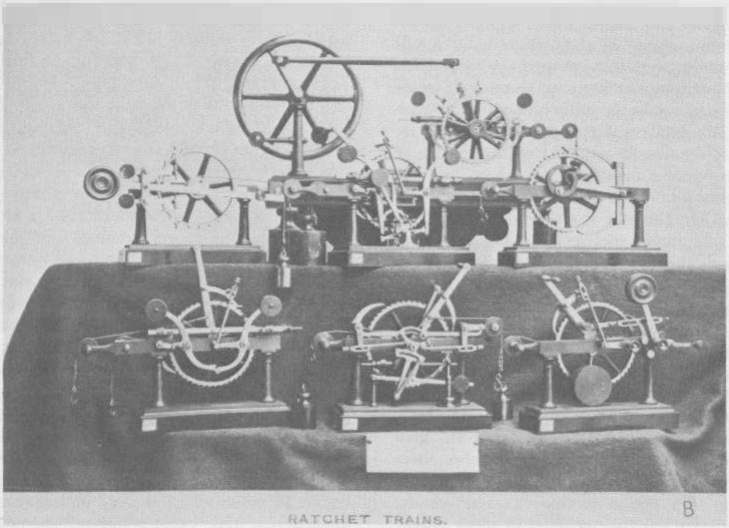


Photo. Science Museum, London

Fig. 7–1. A. Drawing of a ratchet-driven horseless carriage that was built (or proposed) about 1713. B. Display of ratchet

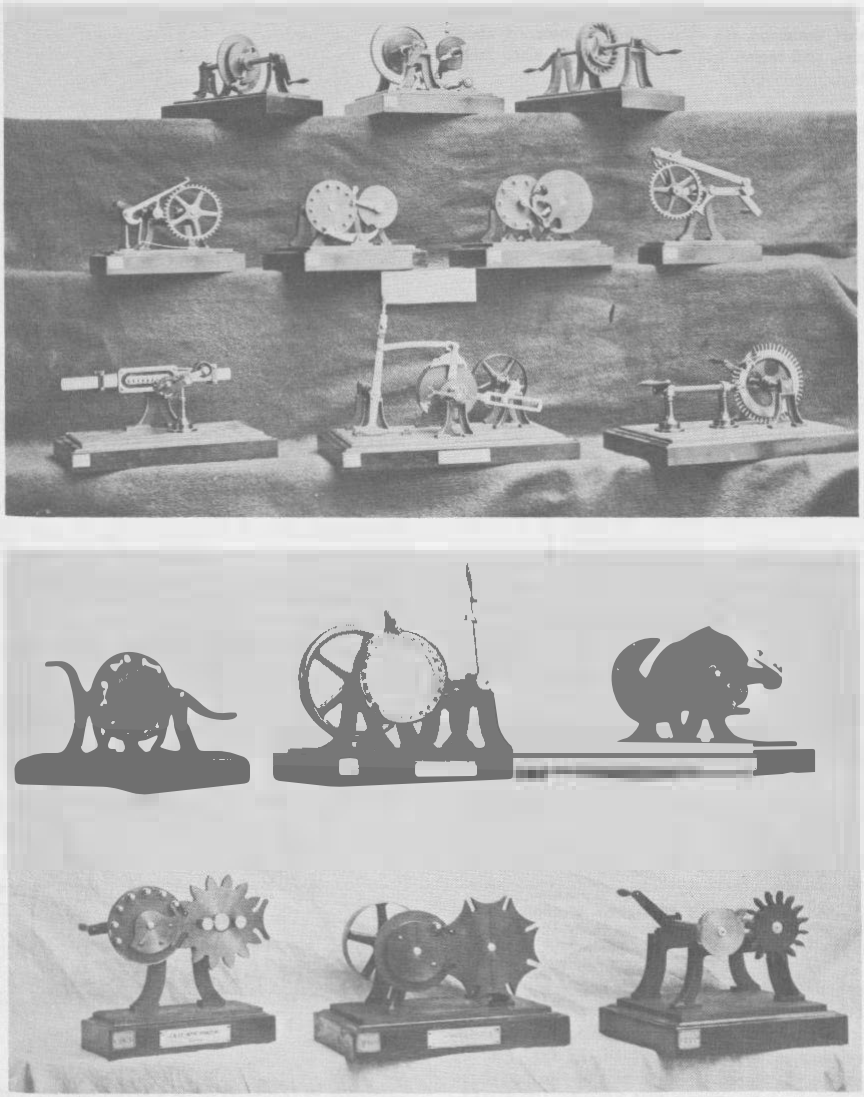


Photo. Science Museum, London

Fig. 7–2. More "ratchet" mechanisms from The Science Museum in London. Note the Genevas, escapements, etc., which are included in this display.

designers than it does at present. The business machine designers are already very concerned about noise, and recent patent literature describes many silent "ratchets."

Ratchet Geometry

Some attention must be given to pawl and wheel-tooth geometries when designing a ratchet. In Chapter 5, Fig. 5-11 shows that it is important to shape ratchet wheel and pawl teeth properly, to reduce impact stress and wear. Proper shape also helps the pawl engage and drive the load. In Fig. 7–6, for example, the pawl is trying to pick up the load of the ratchet wheel. The forces on the pawl as it enters the wheel tooth include a normal force and a friction force, as explained in Chapter 1. The friction force

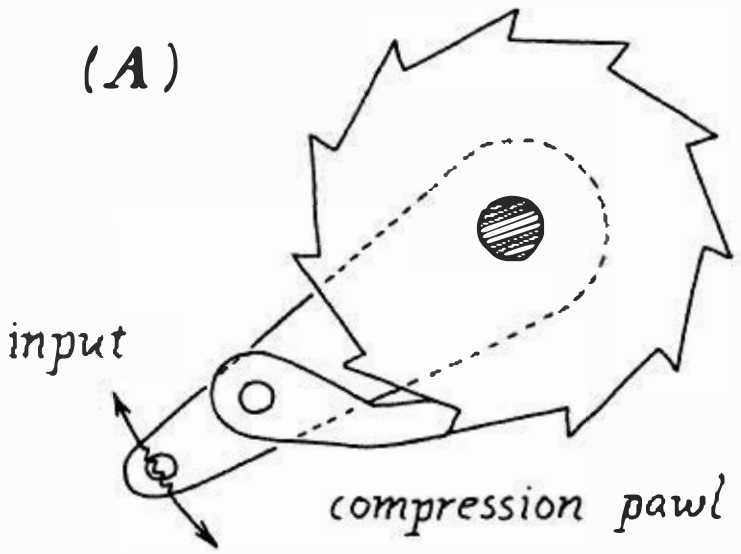


Fig. 7–3. Simple ratchet mechanisms. Example A uses a compression pawl; B uses a tension pawl. Power input is through the pawl in both cases.

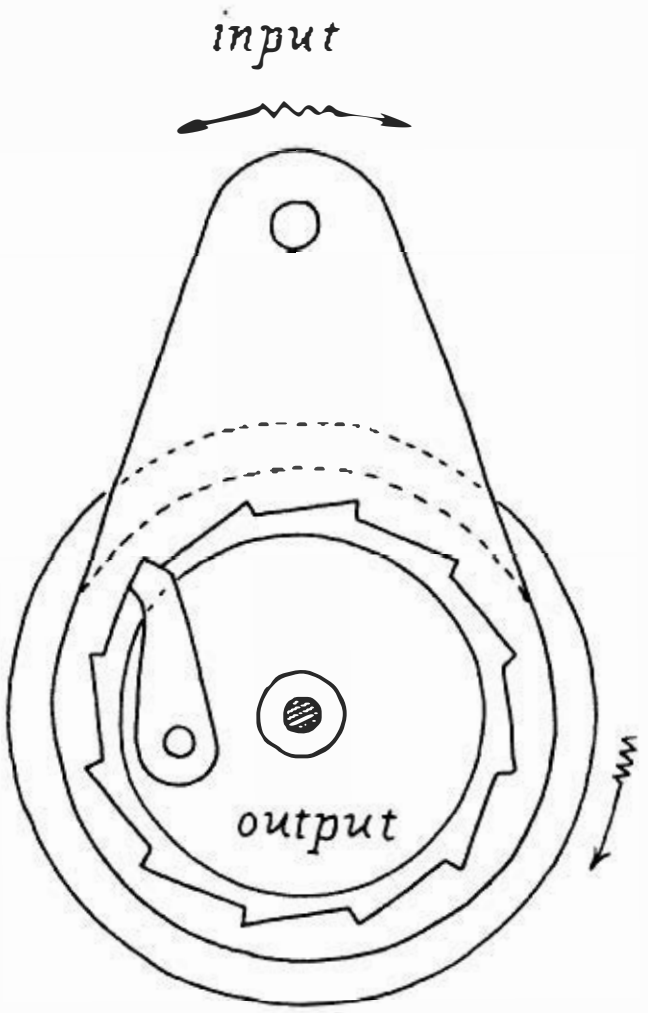


Fig. 7–4. Simple internal ratchet mechanism in which the wheel, rather than the pawl, is the input member.

opposes the *pawl* motion, of course, and so must be drawn in a direction away from the wheel, as shown.

The resultant force on the pawl is in a direction to rotate it clockwise. Yet it must rotate counterclock-

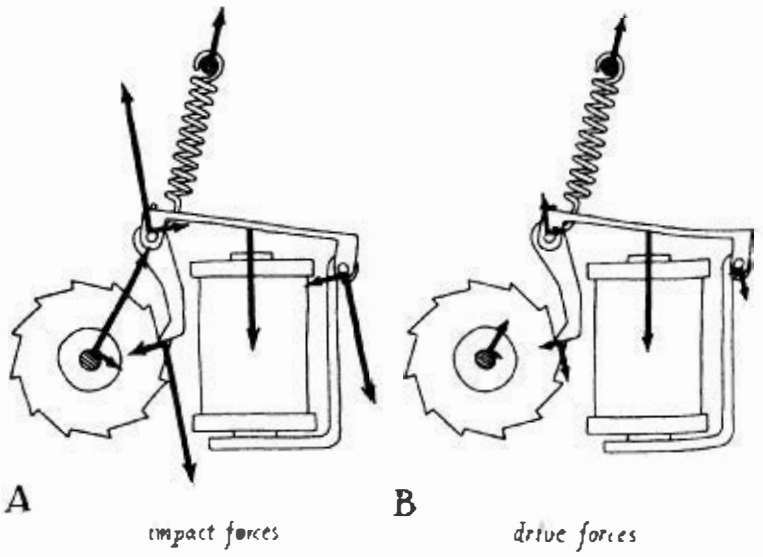


Fig. 7–5. Impact forces (A) on a ratchet mechanism are in the same position and direction as drive forces (B), but are much greater. Forces throughout the entire device are magnified by impact between pawl and ratchet wheel.

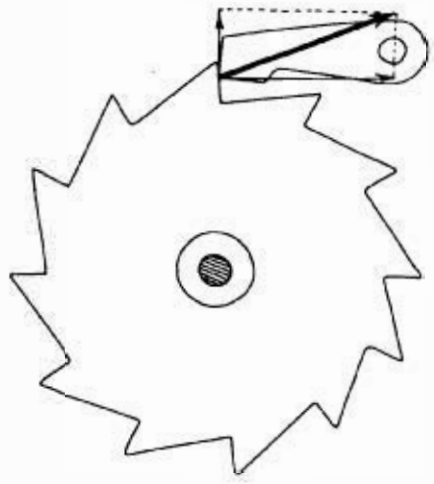


Fig. 7-6. Forces exerted by the ratchet wheel on the pawl as the pawl tries to engage the wheel. The resultant force is in a direction tending toward disengagement.

wise to pick up the wheel tooth. This tendency to disengage must be overcome by providing a spring that urges the pawl against the wheel; or the pawl and wheel teeth must be redesigned to place the resultant pawl force on the correct side of the pawl bearing, as shown in Fig. 7-7. This is usually a better solution to the disengagement problem than a pawl spring, but pawl springs are also required in many applications to get the pawl down into engagement with the wheel more rapidly than it would under its own weight. And, of course, in many designs the ratchet is not oriented in such a way as to engage the drive pawl by weight. Thus, you will see pawl springs even in "correctly" designed ratchets.

Figure 7-8 shows the same ratchet, after full engagement, while driving the load. The forces on the pawl are now those which would be present if the pawl tried to disengage itself from the wheel. Note that the friction force has now reversed, but the normal force remains the same. Note, too, that the counterclockwise torque on the pawl, in this situation, is even greater than it was during engagement. The geometry selected for proper engagement is more than sufficient for continued engagement even if the load should increase.

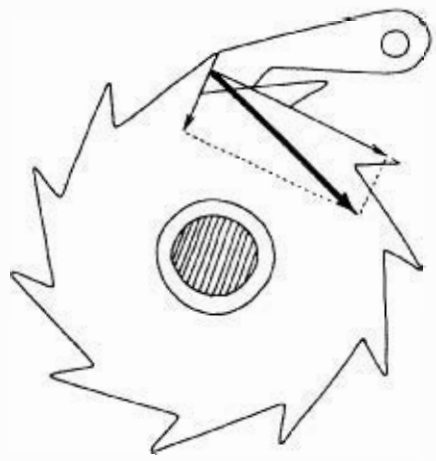


Fig. 7-8. Forces exerted by a ratchet wheel on a fully engaged pawl as the pawl tries to disengage. Note that the friction force has reversed direction from that shown in the previous illustrations.

The location of the pawl bearing with respect to the wheel's bearing should also be given some attention. If the pawl is positioned as shown in A, Fig. 7-9, it will push almost tangentially on the wheel; an efficient arrangement. If the pawl bearing is located at too great an angle with the tangent, it will have to produce a strong cramping force on the wheel to achieve the same useful, or drive force, as shown in B, Fig. 7-9. Of course, it must be positioned at a large enough angle to result in counterclockwise pawl torque during engagement.

Motion Curves

The motion curves of ratchet mechanisms vary widely, depending upon the means used to power

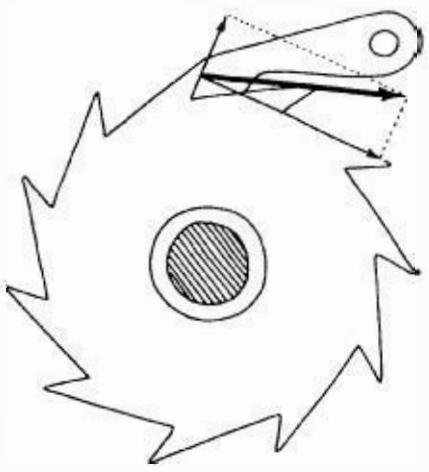


Fig. 7-7. Forces exerted by the ratchet wheel on the pawl as the pawl tries to engage the wheel. With the geometry used in this case, the net force on the pawl is in a direction to encourage engagement.

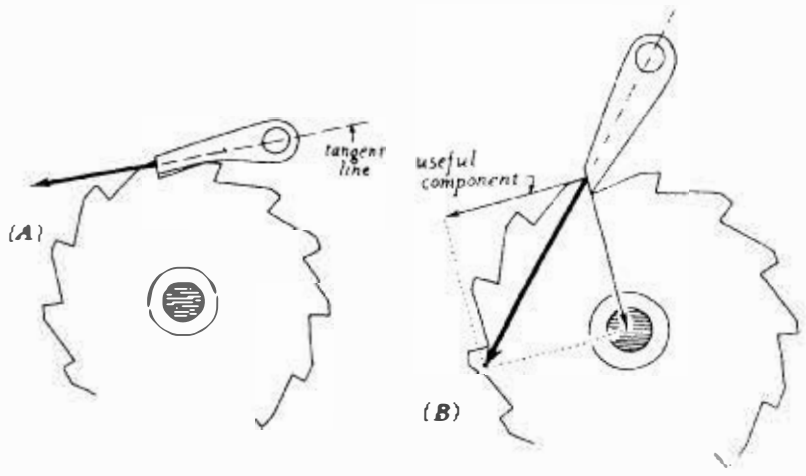
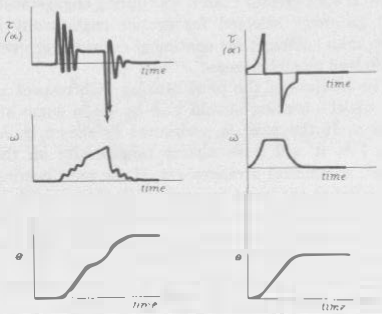


Fig. 7-9. Two pawls that produce the same tangential (useful drive) force. However, the pawl shown in B will tend to cramp the wheel since it is located at too steep an angle to the tangent.

the drive pawl. If a spring, a solenoid, or other pulse [illegible] is used, the torque curve will look [illegible] that plotted in Fig. 4–1. If a cam drive is used, however, the motion curves will be those of Fig. 4–8, or [illegible]. The resultant sets of curves for these two cases are shown in Fig. 7–10. Obviously, the designer should strive for the "smoother" curve set. But he may not have the option of providing a continuously [illegible] ing [illegible]m driver. If he is forced to use the [illegible] or spring drive he is stuck with the impact [illegible]d the

[illegible]

When ratchet speeds increase, certain additional design refinements are almost always required. One



[illegible]

of these is a no-back pawl which prevents the spring-loaded drive pawl from dragging the ratchet wheel backwards as the drive pawl retracts to take another [illegible]. Another, is a non-overthrow de[illegible] of some sort that will prevent the ratchet wheel from moving too far under the [illegible] push (prevent the driver from [illegible] control). If electrical (solenoid [illegible] is used the designer may [illegible] to provide a drive [illegible] which is set by the [illegible] rather than [illegible] the pawl directly, to [illegible] impact and the effects of voltage [illegible]ion.

Now let us [illegible] variety of ratchets. Figures 7–11 through 7–45 give a comprehensive illustrated explanation of ratchet types and the function of their [illegible].

Both [illegible] and no-back pawls in the [illegible] driven ratchet from a large clock mechanism (Fig

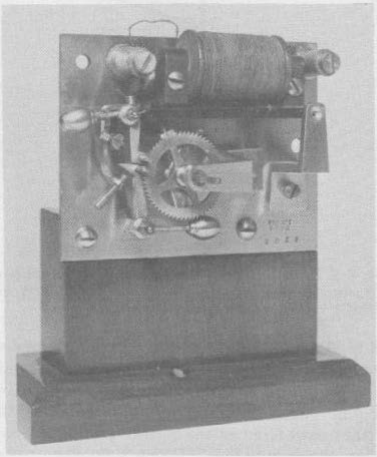


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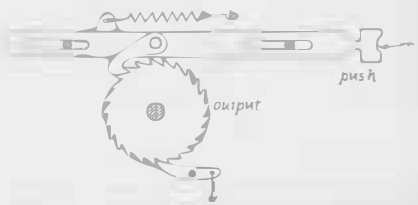
Fig. 7–11 [illegible] driven ratchet from a large clock mechanism.

7–11) are urged against the ratchet wheel by weights [illegible].

In the [illegible] stroke ratchet shown in Fig. 7–12, the distance that the wheel is indexed will depend upon how far the input lever is pushed. This ratchet was used in a vending machine of German manu[illegible].

The wheel of the double-action ratchet illustrated in Fig. 7–13 is indexed during both forward and return strokes of the input crank.

The ratchet in Fig. 7–14 produces unidirectional output for bidirectional rotation of the input shaft. Springs on the arms allow the ratchet teeth to [illegible] engage from the wheel during the return stroke.



[illegible]

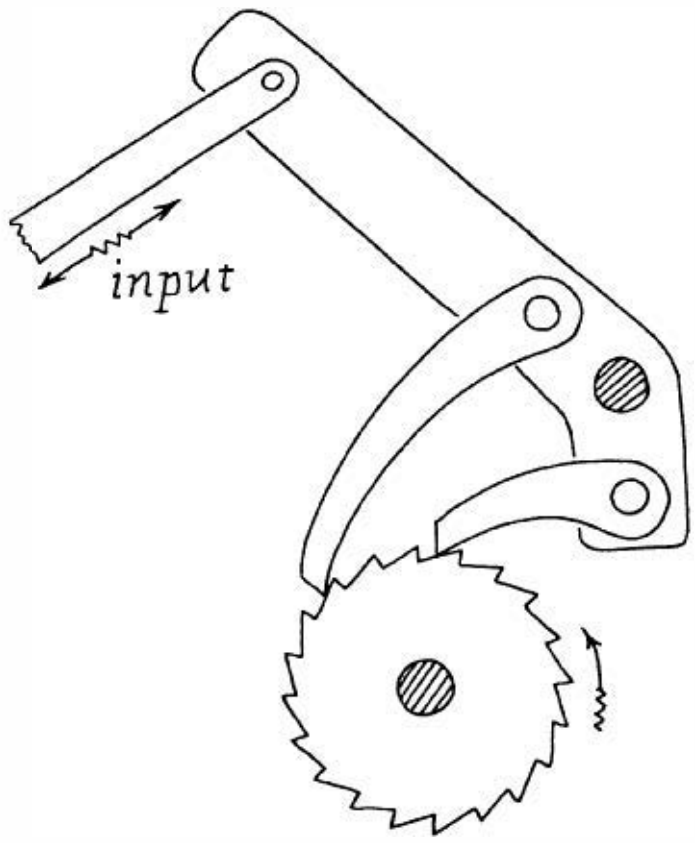


Fig. 7–13. Double action ratchet.

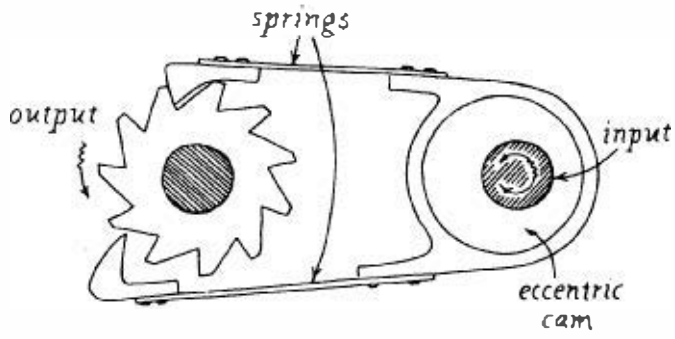


Fig. 7–14. Ratchet producing unidirectional output for bi directional rotation of the input shaft.

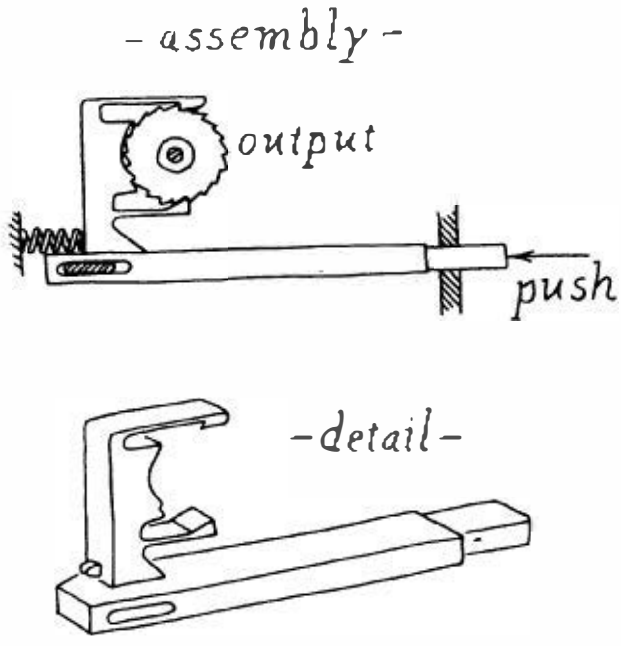


Fig. 7–15. Ratchet with spring-arm drive. (U.S. Patent 3,375,337; Barrett, et al.)

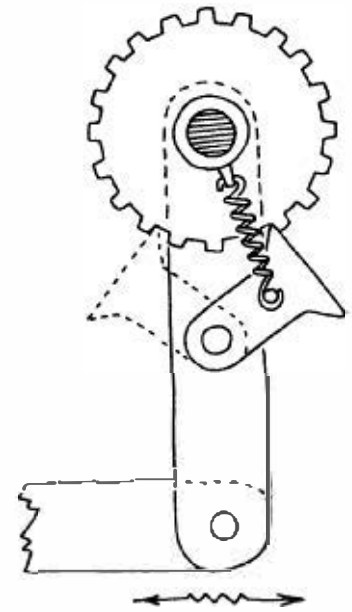


Fig. 7–16. A reversible ratchet.

Note that the top drive tooth operates in compression, while the bottom one operates in tension.

Figure 7–15 shows another ratchet with a spring-arm drive. However, input this time is a push–pull motion rather than continuous rotation.

A reversible ratchet is illustrated in Fig. 7–16. The drive pawl of this mechanism can be flipped from one side to the other to reverse the direction of rotation of the ratchet wheel.

The push-button ratchet mechanism in Fig. 7–17 will produce either clockwise (*CW*), or counterclockwise (*CCW*) motion depending upon which way the toggle has been set.

Another toggle-set bidirectional-output ratchet

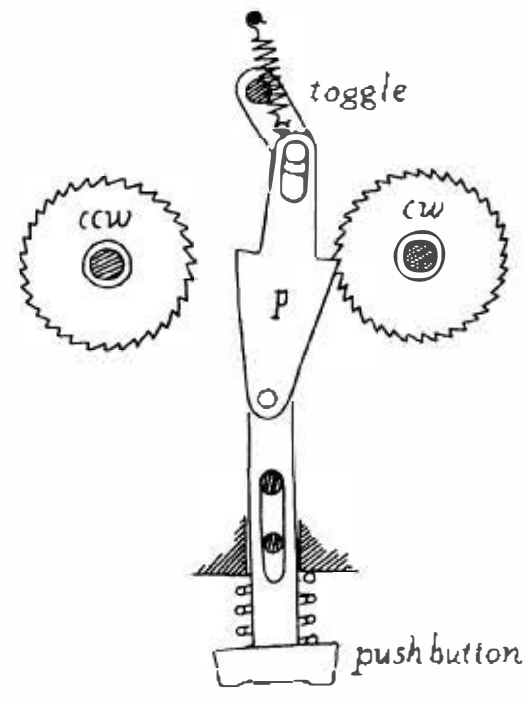


Fig. 7–17. Push-button ratchet mechanism.

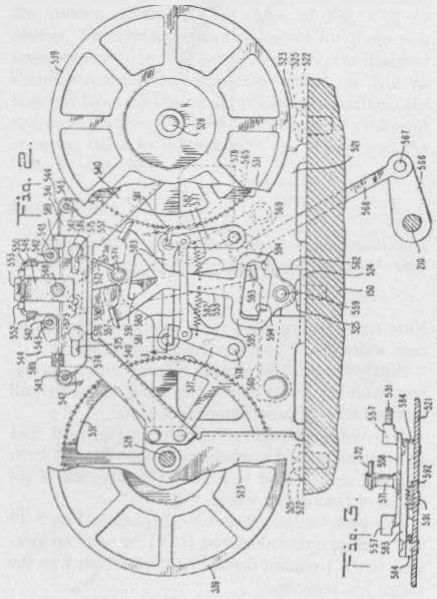


Fig. 7–18. Toggle-set bidirectional-output ratchet mechanism. (U.S. Patent 3,542,183).

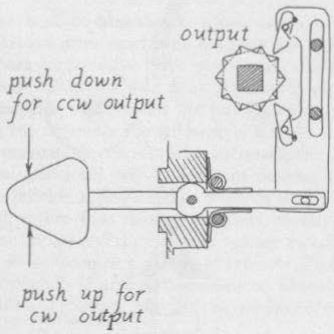
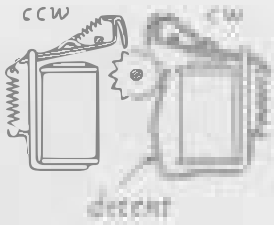


Fig. 7–19. Bidirectional ratchet (U.S. Patent 3,544,748; P. P. Schwab.)



[illegible] bidirectional electrical ratchet. (U.S. Patent 3,473,087, K. J Ylinen.)

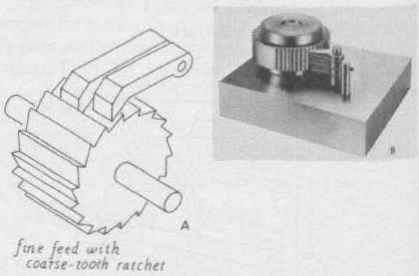
mechanism is shown in Fig. 7–18; this one is used in a contemporary business machine.

The bidirectional ratchet shown in Fig. 7–19, is a mechanism from a manually operated stepping switch.

Figure 7–20 illustrates a bidirectional electrical ratchet.

Multiple-drive pawls mounted on a common shaft can be used to produce fine steps in a coarse-tooth ratchet as shown in Fig. 7–21. The reciprocal motion of the input pawls is less than the pitch of the teeth on the ratchet wheel so that first one pawl engages, and then the others engage, with only one carrying the load at a time. This trick allows us to build fine step ratchets with large, coarse (strong!) pawl and wheel teeth.

Adjustable shields, as shown in Fig. 7–22, can be provided with multiple-pawl ratchets to alter the output stroke. The shield is adjusted to increase or decrease the length of engagement of drive pawls with



Photograph courtesy of The Lowell Corporation

Fig. 7–21. A. Multiple-drive pawls mounted on a common shaft; B. Segmented-pawl ratchet design.

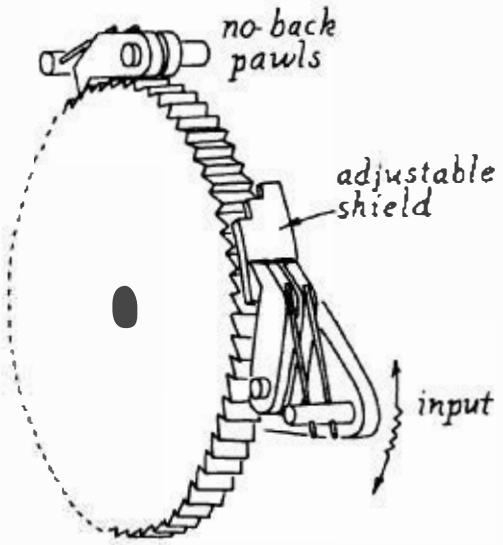


Fig. 7–22. Multiple-pawl ratchets with adjustable shield.

the output wheel. The shield remains stationary during operation of the ratchet, however.

Figure 7–23 illustrates another pawl arrangement designed to obtain fine indexing with coarse teeth. Only one pawl is in full engagement at a time.

In this two-speed ratchet (Fig 7–24), the two pawls move together on a common input shaft. The right-hand pawl engages the right-hand ratchet wheel with every input stroke, advancing this wheel and driving the "fast" output shaft. The left-hand ratchet wheel has a smaller diameter than the right-hand one, thus the left-hand pawl does not normally engage this second wheel. Twice per revolution of the right-hand wheel, however, the input pawls encounter an extra-deep ratchet tooth *d*. This allows

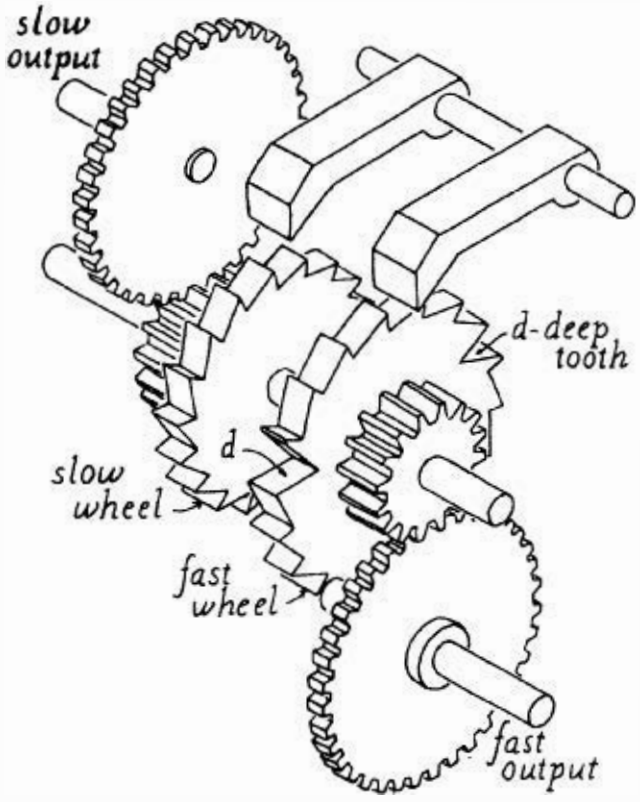


Fig. 7–24. Two-speed ratchet.

the drive pawl to drive down to a smaller diameter and the left-hand pawl engages the left-hand, or slow, ratchet wheel, producing output on the "slow" shaft.

Ratchet wheel teeth need not be saw-tooth shaped. In this hydraulically actuated mechanism of Fig. 7–25, a square block serves as a ratchet wheel. A

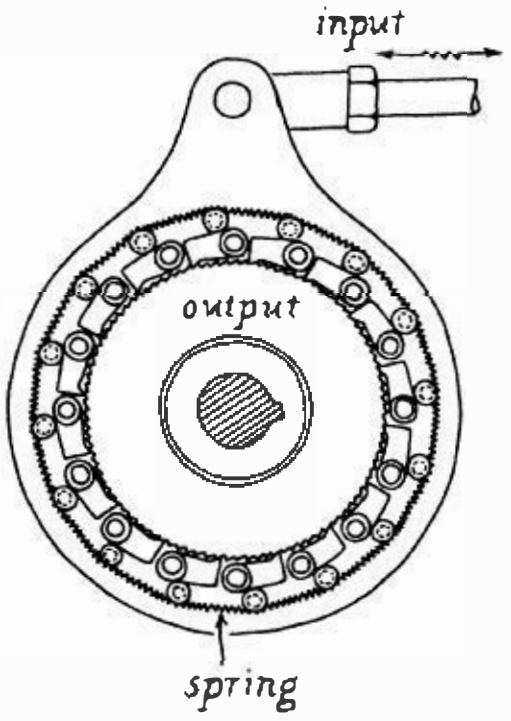


Fig. 7–23. Another multiple pawl design. (U.S. Patent 3,505,890; C. G. Peterson.)

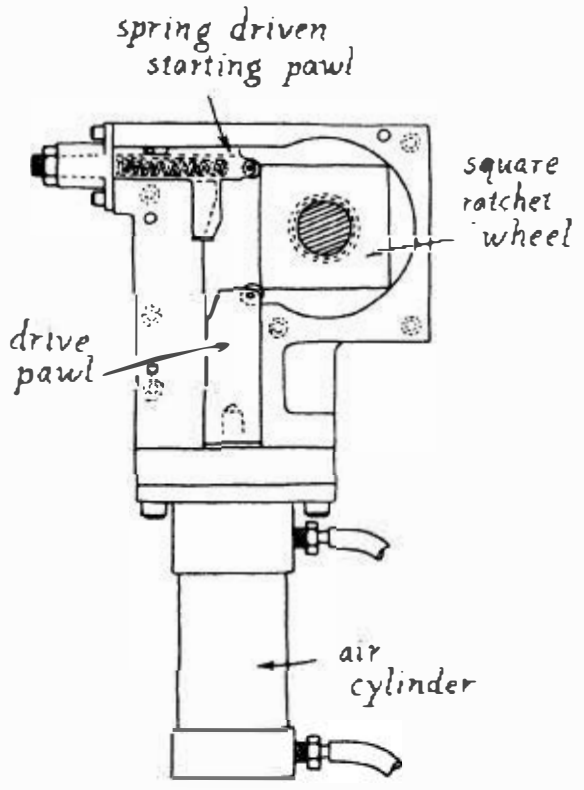


Fig. 7–25. Mechanism with square block ratchet wheel. (U.S. Patent 3,548,684; A. R. Gregersen.)

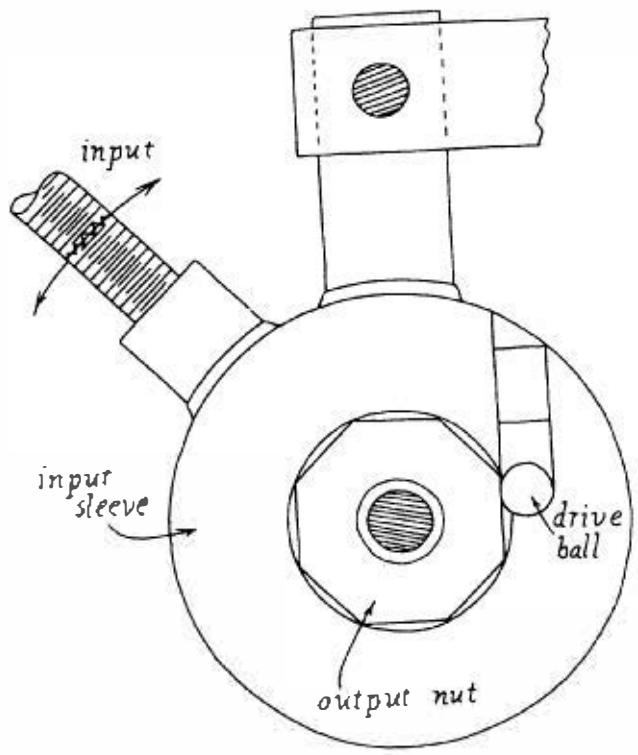


Fig. 7–26. Odd-shaped ratchet wheel. (U.S. Patent 3,323,383; V. A. Parsley.)

spring-loaded pawl serves to start output wheel motion so that the drive ratchet can keep driving a different face of the block with each stroke.

In Fig. 7–26 is another odd-shaped ratchet wheel; really a one-way ball clutch. Here the hexagonal nut is the output wheel. The motion of the input crank in one direction will produce no output motion as the nut merely moves the drive ball out of the way. Motion in the reverse direction, however, will produce output because the ball will jam between the nut and the input sleeve.

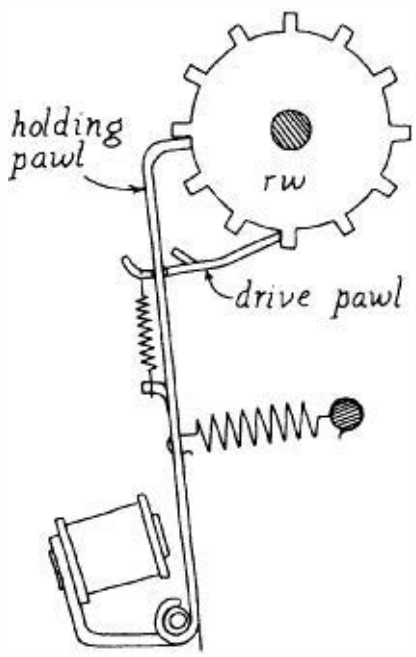
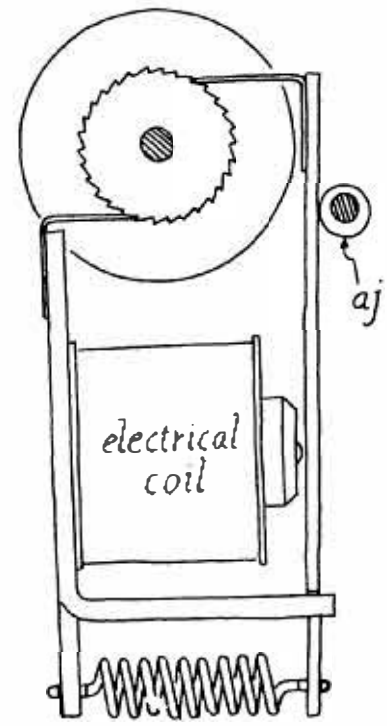


Fig. 7–27. Basic, low-cost solenoid ratchet from a Japanese counting mechanism.



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 7–28. Another low-cost solenoid ratchet.

A basic, and low-cost, solenoid actuated ratchet from a Japanese counting mechanism is shown in Fig. 7–27. Drive arms, drive pawl, and holding pawl are all stamped from sheet metal.

Another low-cost electrical ratchet, though not quite as simplified as the first, is seen in Fig. 7–28. Flat springs are used for drive and holding pawls. Note that an eccentric adjustment (*aj*) has been provided to control the starting position of the drive pawl (and, not incidentally, the de-energized air gap of the drive solenoid).

If a ratchet's pawl and drive arm can be made light enough and the teeth on the ratchet wheel can be made very fine, then a ratchet can be operated at very high speeds. The sketch in Fig. 7–29 shows a very-low-cost ratchet motor that is used to power toy racing cars. An electrical coil is energized with 60-cycle AC power. The ratchet armature, and pawl (*B*), are merely pieces of spring steel which are

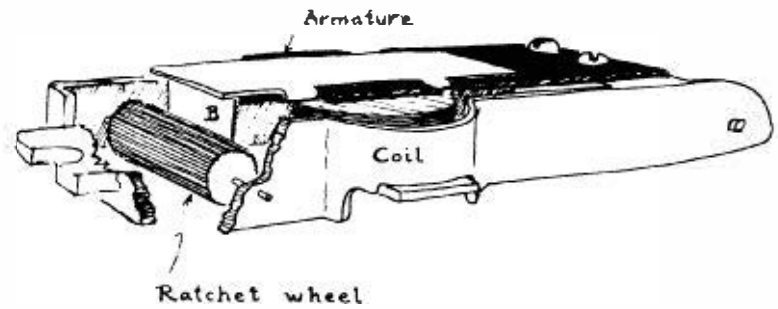
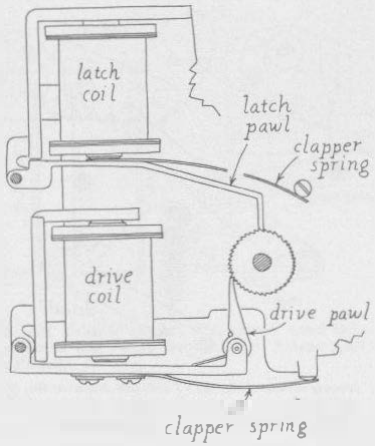
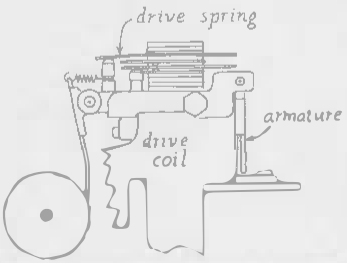


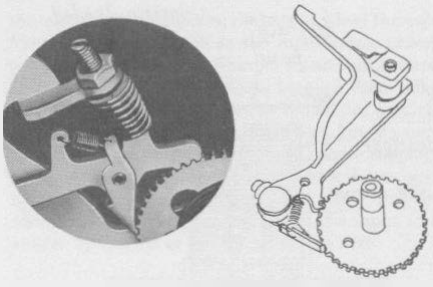
Fig. 7–29. Low-cost ratchet motor used to power toy cars.



attracted to the coil every time it is energized and which move away from the coil under their own spring power whenever the current level in the coil nears zero. The armature, therefore, vibrates at a 60-cycle rate. The pawl (*B*), engages the ratchet wheel, driving it (and the racing car) in a forward direction. A no-back pawl (not shown) prevents the ratchet wheel from reversing when the armature moves upwards.

Solenoid operated ratchets have been used for many years in high-performance stepping switches for telephone systems, etc. There are two basic kinds: direct drive and spring drive. The latch coil at the top of the direct-drive model shown here in Fig. 7-30, is energized to remove the latch pawl from



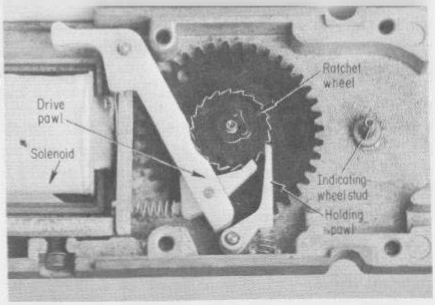


Photograph courtesy of the GT E Automatic Electric Company

arm and ratchet from a high-performance stepping switch.

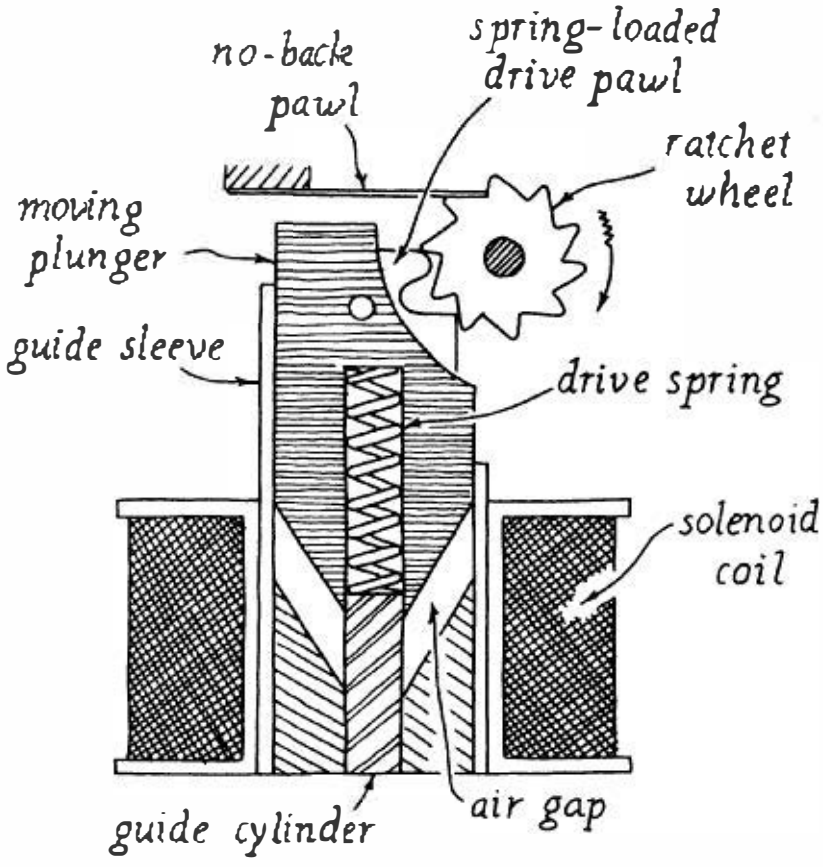
the ratchet wheel teeth. The coil is then energized to index the ratchet wheel. Clapper springs return both pawls to their de-energized condition to complete the stroke. (Drawing made from information supplied by C. P. Clare and Company.)

Figure 7-31 illustrates a more popular approach for a telephone stepping-switch ratchet. The drive coil moves the drive pawl upward, loading a flat drive spring. When the coil is de-energized, the drive spring moves the pawl down to the position shown, indexing the ratchet wheel. Other circuits in more uniform the switching system of Fig. 1- voltage variations, magnetic aging, etc., alter the amount of energy delivered to the ratchet wheel. Non-overthrow and no-back control is provided by which are not shown in the illustration.



Photograph courtesy of the Veeder-Root Company and Magazine, Dec 23, 1965 p 121 ff

Fig. 7-33 Ratchet-drive mechanism of high-performance



Drawing courtesy of The Heinemann Electric Company

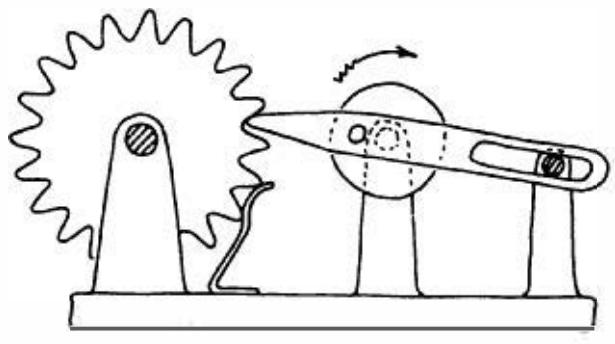
Fig. 7-34. Ratchet mechanism for an electro-mechanical stepping motor.

tion. (Drawing made from information supplied by C. P. Clare and Company.)

The drive arm and ratchet wheel from a high-performance stepping switch is shown in detail and photographed, in Fig. 7–32. These are good illustrations of a stepping-switch drive arm showing a spring-loaded drive pawl and a fixed non-overthrow tooth.

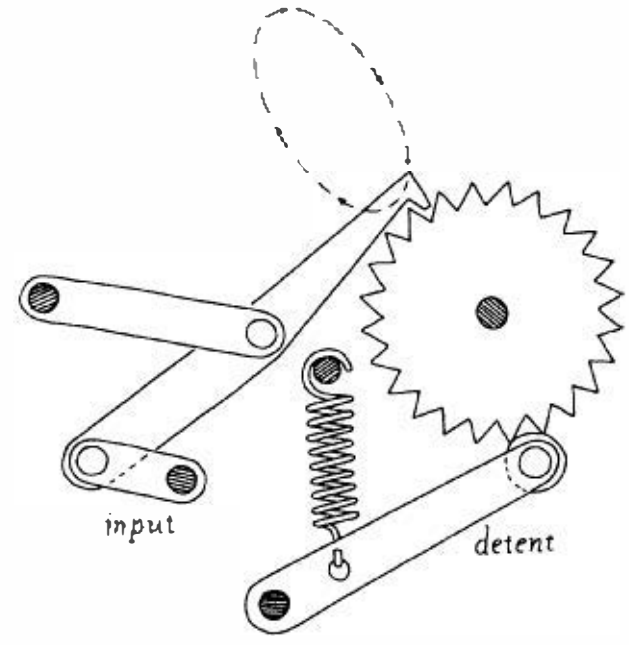
This photograph (Fig. 7–33) of the ratchet drive mechanism for a high-performance electro-magnetic counter shows the drive pawl, ratchet wheel, holding pawl, and drive solenoid.

Figure 7–34 is the ratchet mechanism used in an



Drawing courtesy PRODUCT ENGINEERING Magazine; Oct. 26, 1964; pp. 108, 110

Fig. 7–35. Mechanism with drive arm driven by slider crank device.



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 181 ff

Fig. 7–36. Linkage-driven ratchet mechanism.

electro-mechanical stepping motor. The drive pawl also serves as a non-overthrow pawl. (See also Fig. 14–28.)

A mechanism in which the drive arm is driven by a slider crank device is shown in Fig. 7–35. A linkage-driven ratchet usually exhibits a better acceleration-versus-time curve than does an impulse ratchet.

Another linkage-driven ratchet mechanism can be seen in Fig 7–36. Note that a spring-loaded detent has been provided to serve the functions of both non-overthrow teeth and no-back pawl. This is only possible at low speeds.

This time, in Fig. 7–37, there is a third linkage-operated ratchet producing two outputs; one in a large wheel and one in a small wheel.

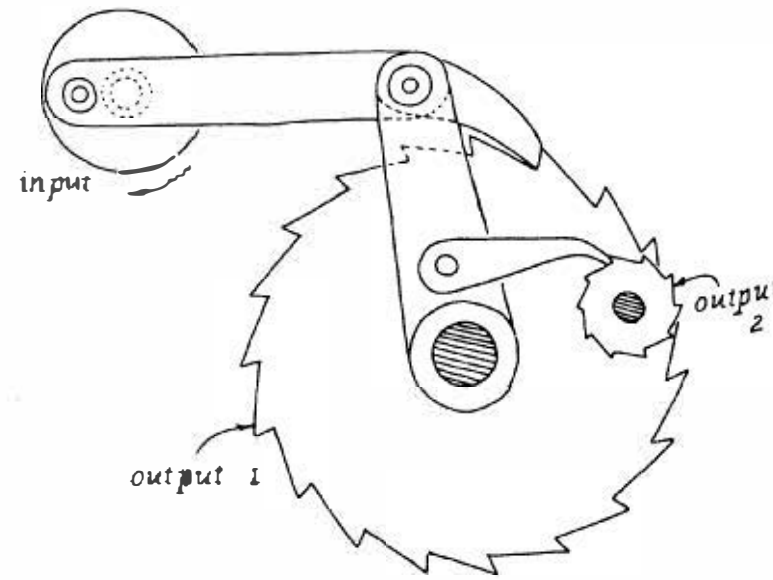


Fig. 7–37. Linkage-driven ratchet mechanism.

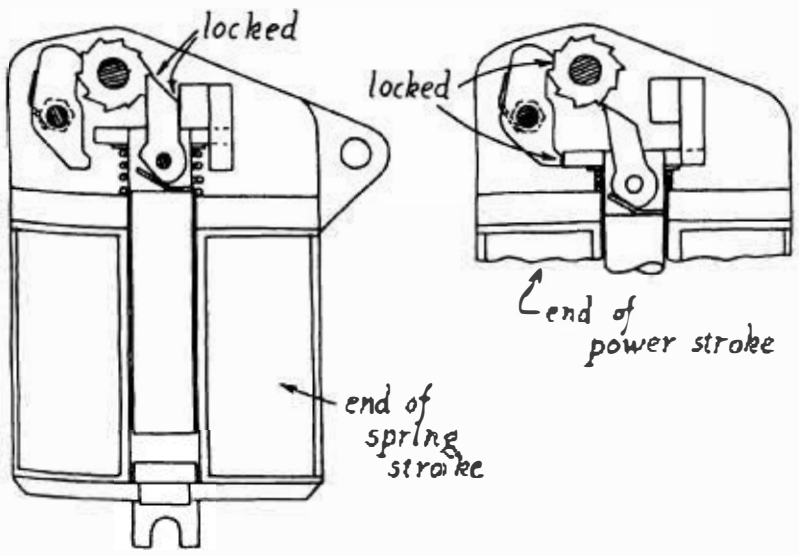


Fig. 7-38. Solenoid ratchet with output wheel locked at end of power and return strokes. (U.S. Patent 3,501,968; G. D. Fredell.)

An electrically actuated ratchet in which the output wheel is locked at the end of both power and return strokes is seen in Fig. 7-38. People are always struggling to improve the stability and control of a ratchet mechanism! But there are still times during which the wheel is virtually on its own.

In Fig. 7-39, is a pneumatically actuated ratchet where air is exhausted from the latch piston cylinder to extract the latch. Air is then introduced to the index cylinder to move the ratchet forward.

With a toothless, or silent ratchet, as in Fig. 7-40, a toggle mechanism jams a drive link down against the output wheel, grabbing the output wheel through friction and indexing it as the input crank moves toward the left. Moving the input crank back toward the right will eliminate the jam between drive links and wheel and allow the arm to retract for another bite. The device depends upon the high forces generated by the toggle mechanism to eliminate slipping. Many versions of this type of device have been built. Rest positions of the output wheel are determined by the length of stroke of the input arm and/or by detents or brakes not shown here.

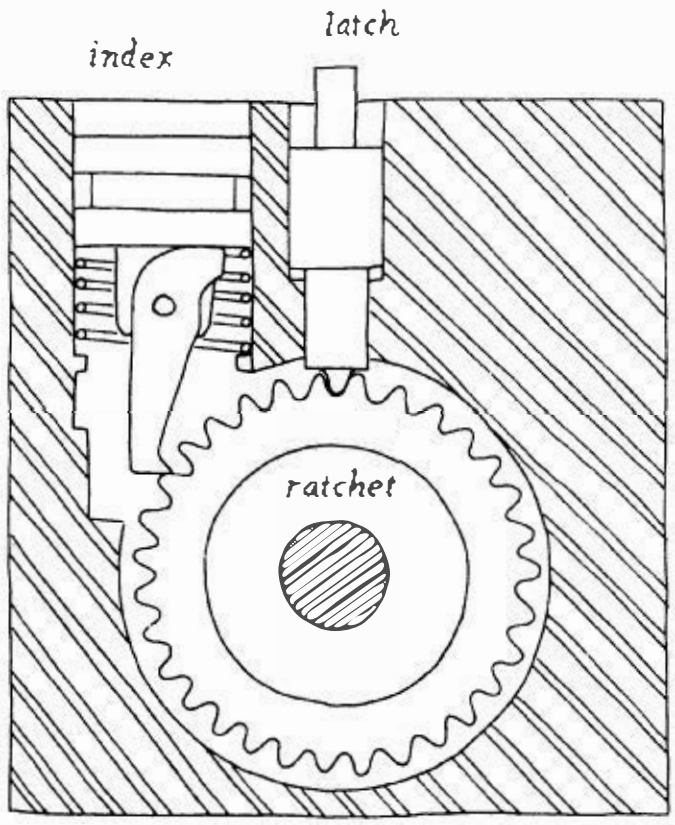
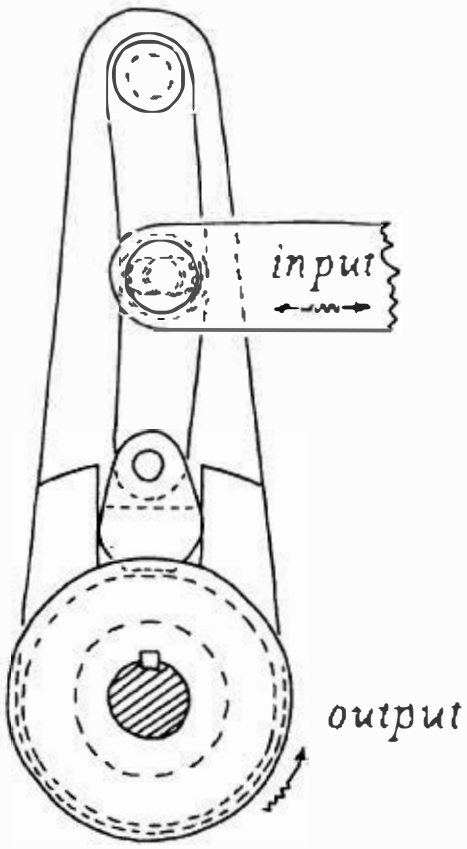


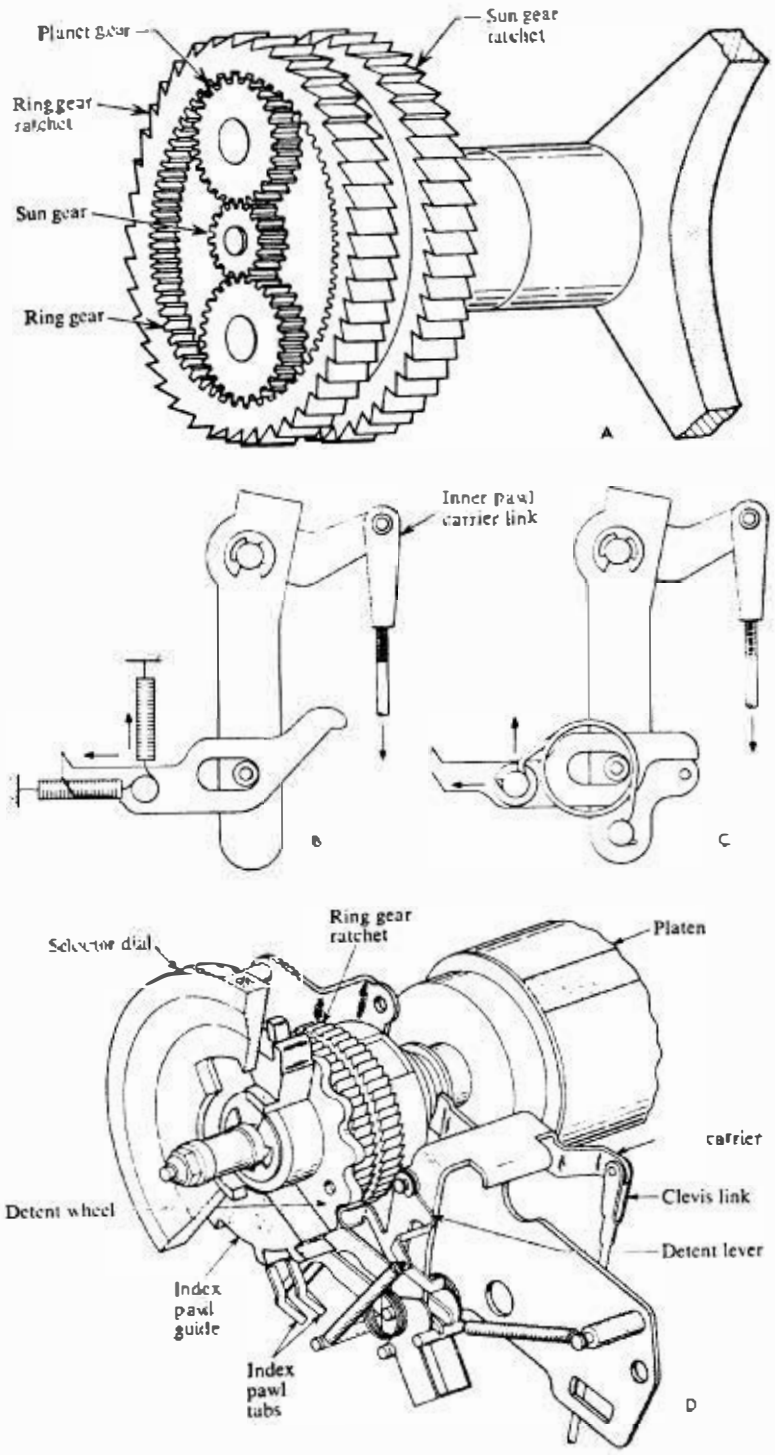
Fig. 7-39. Pneumatically actuated ratchet.



By permission from Douglas Greenwood, PRODUCT ENGINEERING DESIGN MANUAL; New York: Morgan-Grampian Inc., 1038

Fig. 7-40. Toothless, or silent ratchet.

Figure 7-41 shows a multiple step index mechanism that was designed for the IBM Selectric Composer. This machine must handle 15 different sizes of type, ranging from 5 to 20 points. This means that 15 different line spacings have to be provided by the platen indexing mechanism. IBM's answer is this sophisticated ratchet mechanism that can be adjusted to provide 15 different output motions. We cannot go into all the details of this impressive device, but in general it works as follows: two platen indexing ratchets are provided. Their outputs are combined by a differential gear system. Figure 7-41A shows the two ratchet wheels and the gears that are used to add their motions. A cage (not shown) supporting



Drawings courtesy of IBM; J. Res. Dev., Vol. 12 No. 1; R. D. Mathews, Jan. 1968; pp. 76-85

Fig. 7–41. Multiple step index mechanism. A sophisticated ratchet.

the planet gears is connected to and drives the platen.

A differential of this kind will produce three different outputs if the two input ratchets have fixed length strokes. The length of the output stroke will depend upon whether or not the left ratchet, or the right ratchet, or both are being driven at a given time. Additional variation in output motion is obtained by varying the strokes of the two pawls that drive the ratchet wheels. This is accomplished by providing an adjustable stop, called the "index pawl guide" in Fig. 7-41D. The drive pawls themselves (best seen in Fig. 7–41C, and schematically in Fig. 7–41B) are always moved through the same input motion, but are connected to their input arms by rollers that are free to move in slots. When the tip of a pawl encounters a stop, therefore, the pawl comes to rest and the remainder of the input motion merely loads a spring. By adjusting the stop position IBM engineers were able to provide several different stroke lengths for each drive pawl.

An adjustable stop of this type is one of two basic methods commonly used to vary the stroke of a ratchet. The other technique, an adjustable shield arrangement, is shown in Fig. 7–22. In each case the pawl arm moves through a stroke whose length never varies; the active length of the stroke is varied by an adjustable shield or an adjustable stop.

As a matter of fact, both techniques are used in the ratchet mechanism of Fig. 7-41. Although it is not shown in the illustration IBM also used an adjustable shield to provide a "zero index" (no platen motion) option for the operator.

Figure 7–42 shows another slider-crank ratchet. A simple and low-cost arrangement that minimizes impact forces in the same way as a four-bar linkage. This sketch was made from an all-plastic design.

Depending on the setting of the two control levers the commercially available device shown in Fig. 7-43 will ratchet in a clockwise direction, a counter-clockwise direction, or neither. When "neither," the output will either be locked or completely free.

There is no metal-to-metal contact between drive teeth or detent and the output wheel in this magnetic

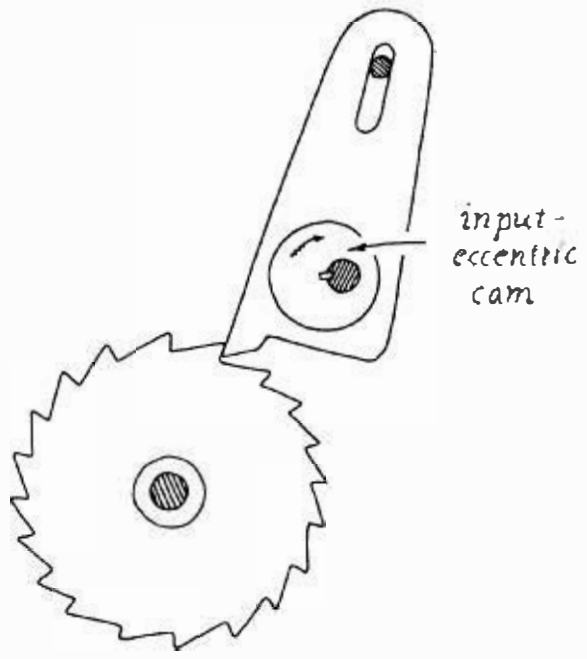
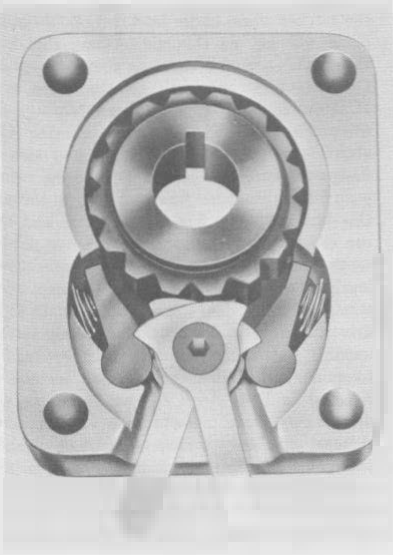


Fig. 7–42. Slider-crank ratchet.

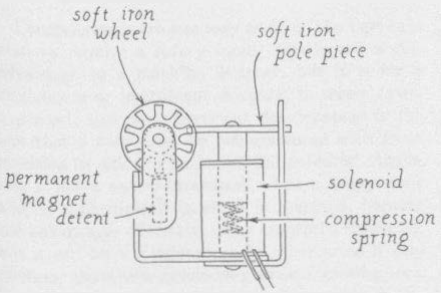


Photograph courtesy of The Lovell Corporation

Fig.7 -43. Reversible orfree-running ratchet.

ratchet, Fig 7–44. Instead, magnetic attract[ion] used for both drive and detent[s].

When one of the coils shown in Fig 7–45 is energized with 60-cycle p[ower] will [oscillate] a rubber ball "spring" in the b[ottom] of the [structu]re, and the ratchet paw l will also oscillate. The [output] will step continuously at 60-cycle rate,



Drawing courtesy of MACHINE DESIGN Magazine; Dec [illegible] p. 121 ff

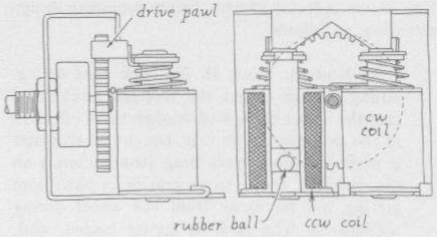
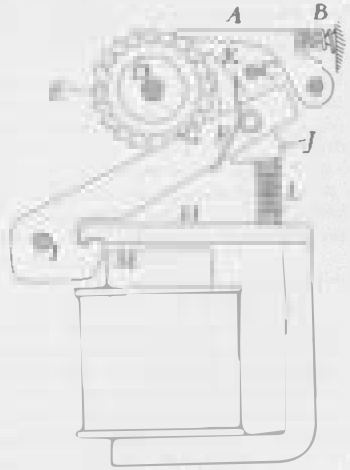


Fig. 7–45. Oscillating ratch[et]. Patent 3,530,346 R. Mariani.)

[coil] turned [clock]wise and counter-clockwise [output] is avai[lable].

Anatomy of a Ratchet

I[n this] and the chapters which follow, each dis[cussion] of a particular type of [intermittent] mechanism will be summarized by an illustration of a typical example and a point-by-point discussion of its "anatomy." Part of the idea is to show that the design of a successful mechanism involves more than the d[imensional] of its general configuration. Many details— materia[ls], [bear]ings, the size of [bear]ings, etc, must be accurately accounted for. As a first example, consider the ratchet shown in Fig 7–46.



Drawing courtesy of MACHINE DESIGN Magazine; Dec [illegible] p. 121 ff

The letters indicate various points requiring design attention, as follows:

A. No-back pawl; B. No-back pawl spring, strong enough to get the no-back pawl down into the wheel teeth soon enough to be effective at the maximum cycle rate, but not so stiff that it produces unnecessary drag (friction load) on the wheel; C. Stop—to prevent drive pawl from getting too far away from the wheel during withdrawal. The pawl is spring loaded, and, just as any spring-mass system, will overrespond to shock inputs (which occur when it is suddenly pulled back over the wheel teeth); D. Ratchet-wheel bearing—small diameter to reduce frictional drag torque, especially during impact load periods, but the diameter is large enough to keep bearing stresses reasonable for long life. This is usually a principal wear point; E. Non-overthrow teeth—engage the tooth on the drive arm (G) to prevent the wheel from turning more than it should in a given drive cycle. These non-overthrow surfaces generally take much abuse; F. Drive-pawl spring—the comments made on the no-back pawl spring (B) apply here also. In addition, this spring must be strong enough to force the drive pawl to engage the ratchet wheel under load if tooth geometry alone will not do this. Sometimes, the drive spring (L) can also serve as the pawl spring; G. Non-overthrow tooth on the drive arm; H. Iron clapper; I. Drive-arm bearing—the comments made for the wheel bearing (D) above, also apply here. It is a good idea to design the magnetic circuit so that the return path for the magnetic flux does not run through this bearing, since flux means force, and this can cause cramping and wear; J. Drive pawl—tooth properly shaped to reduce impact stress levels. Hardened; K. Drive pawl—starts drive stroke about one-half tooth, back of tooth to be driven next. Less backlash would help reduce impact, but would mean tighter production tolerances; L. Drive spring—used instead of solenoid to limit drive torque and make the drive forces more uniform with varying input voltage, etc. Drive torque must be nearly balanced by friction and inertial loads in the ratchet wheel and in related parts of the machine. The inertia ratio between driver and load affects impact levels, bounce of arm, and stability as discussed in Chapter 5; M. "Air gap"—will affect initial magnetic forces on the drive arm and, therefore, the indexing rate.

CHAPTER 8

Cams

Advantages and Disadvantages

Cams, like ratchets, are very popular intermittent motion devices; but the two are rarely competitors for the same jobs. Like ratchets, cams are very simple mechanisms, at least they involve very few parts. Unlike ratchets, however, cams give the designer great freedom in his choice of motion patterns. Acceleration, velocity, and displacement can all be manipulated and controlled by the cam designer.

Other general advantages which can be obtained with cam systems are such things as:

a. High load-carrying ability
b. Low shock and acceleration—virtual elimination of impact
c. Very long life
d. High reliability
e. Quiet operation.

Disadvantages are less easy to find. The best cam systems require a rotary input; this is not a disadvantage to a machine designer, but it is for a mechanism or instrument designer, in many cases. A second, and more pertinent disadvantage is the fact that a cam must be manufactured with great precision in order to realize its full potential. Again, this is not a serious drawback when a heavy duty and/or sophisticated machine is involved, because the advantages offered by cams are worth their cost, but it can be a disadvantage in other areas if production quantities preclude precision casting, etc. Cam shapes are complex and good cams are costly. But many companies make them and superb manufacturing equipment is readily available.

Driver and Driven Members

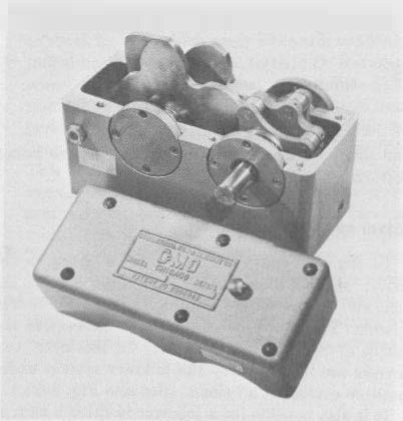
In the best cam systems, the cam, itself, is the driver and a reciprocating or pivoted arm is the follower. Figure 8–1 is an example of such a cam system. The lobed member is the driver and the group of rollers is the follower. In this case, two drivers are used to keep the follower system under positive control at all times. (See also Fig. 8–23.)

It is also possible for a follower to drive a cam as in Fig. 8–2. Generally speaking, this class of cam systems is less desirable than the first, as far as acceleration patterns, etc., are concerned, but they find many applications. I think, for example, that the Geneva might be considered as a cam system of this class: but it is so important as an indexing device that we will devote a separate chapter to it later. The "inverse escapement" is also a follower-driven cam system. This mechanism is covered at length in the chapter on escapements.

Machine design handbooks often discuss the pressure angle between a cam and its follower. Such treatments are always related to the first class of cam systems in which the cam drives the follower. If the follower is mounted in slide bearings it is essential that the resultant forces generated by the cam on the follower deviate as little as possible from the axis of the follower. 30 degrees is usually considered a maximum pressure angle in a practical

situation. (See Fig. 8-3a.) If the follower is [illegible] as in Fig. 8-13, for example, a pressure angle of 45 to 55 degrees is considered acceptable. In an indexing cam, as in Fig. 8-7, 60 degrees is not [illegible]

In the second class of cam systems, however, where the follower is doing the driving, it is desirable to have a very large pressure angle as shown in Fig 8-3b, so that a small force in the follower will produce a large torque on the cam. 60 to 80 degrees would be very desirable; though 45 to 60 degrees is more common. This large angle gives the follower the mechanical advantage

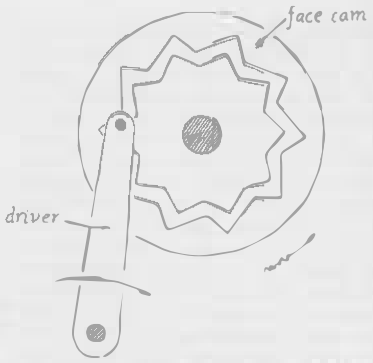


Photograph courtesy of the Commercial Cam and Machine Company

Fig. 8-1. In this modern indexing cam system the input is through the lobed cam on the left. The roller wheel, on the right, moves intermittently. The device shown in the illustration will produce one or two stops per revolution of the input, depending upon the shape of the drive cam. (See also Fig. 8-23.)

[illegible]

Because the cam designer has a lot of [illegible] acceleration, velocity, and displacement patterns when designing a cam, he has traditionally given this area a great deal of thought and attention. In fact, he is probably ahead of all other types of [illegible] designers in this respect, since they are generally required to accept the patterns inherent in [illegible], whether they are ratchets, stepping



Drawing courtesy of MACHINE DESIGN Magazine; [illegible] *1, 1966; pp 86—93*

[illegible] this can be used in relatively light-duty applications.

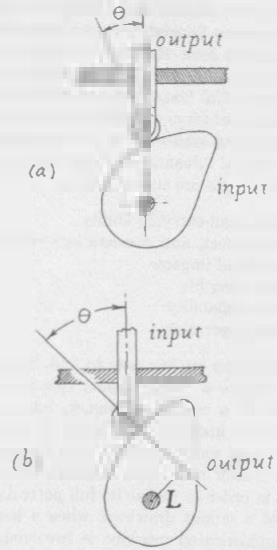


Fig. 8-3. Pressure-angle requirement for two types of cam system. In sketch (a) the cam is driving a sliding follower and the pressure angle θ, must be kept less than 30 degrees to avoid cramping the follower in its support plate. In sketch (b) the pressure angle θ should be kept large to maximize lever arm L, and produce a large torque on the cam, since the follower in this situation is driving the cam.

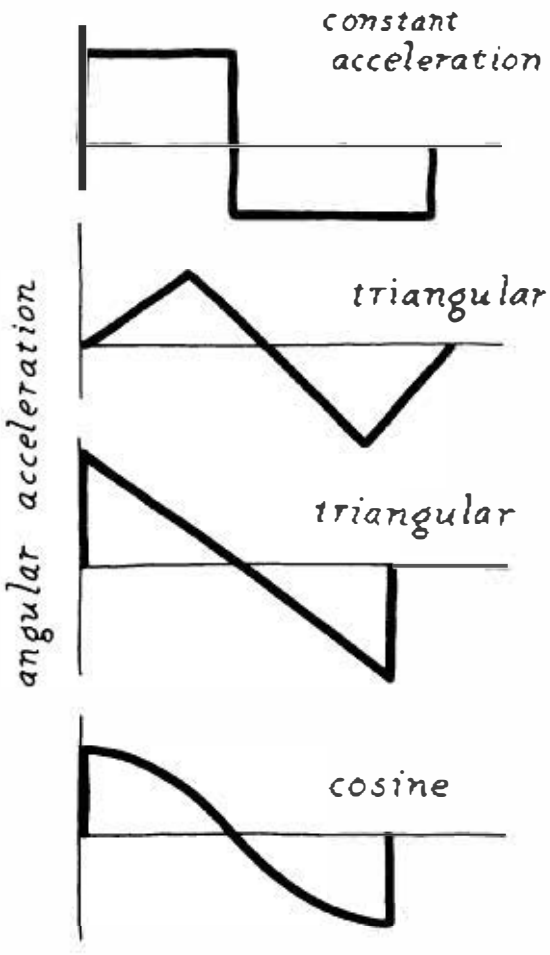


Fig. 8–4. Unpopular acceleration patterns for indexing cams.

motors, escapements, etc. Designers of cams for heavy-duty machinery or high-speed business machines, especially, have spent a good part of their professional life thinking about the "ideal cam profile." The results are described at length in many excellent books on cam design and cannot be repeated here as space does not permit. But some general concepts can be considered.

One historical goal of the cam designer has been to avoid instantaneous change in acceleration of the output, because this means the instantaneous application of force; and "instantaneous force" means shock and oscillation, as explained in Chapter 3. Thus, the cam designer tries to avoid acceleration time patterns such as those of Fig. 8-4, and looks, instead, for smoothly changing patterns. It is virtually impossible for the cam designer to build a practical cam with zero slope at the beginning and end of motion, but he can succeed in making finite slopes such as those shown in Fig. 8–5. Designers say, however, that since a truly zero slope at the beginning and end of motion is impossible in a practical cam, they look instead, for "continuity of acceleration."

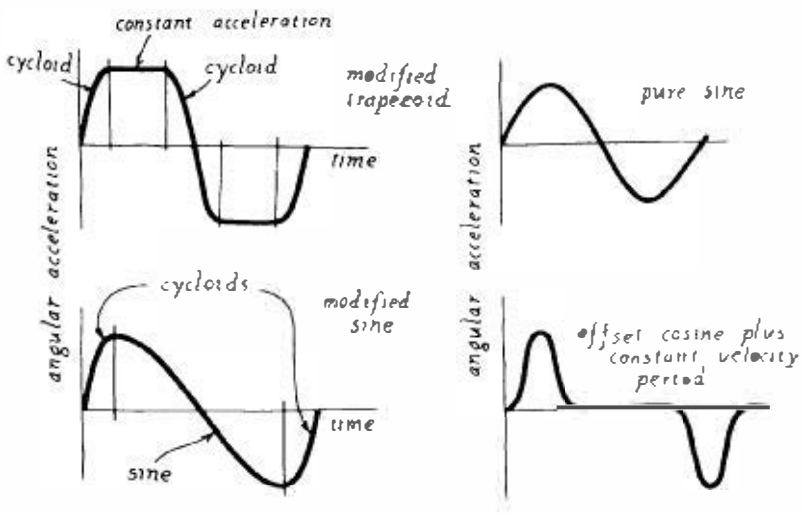


Fig. 8–5. Popular acceleration patterns.

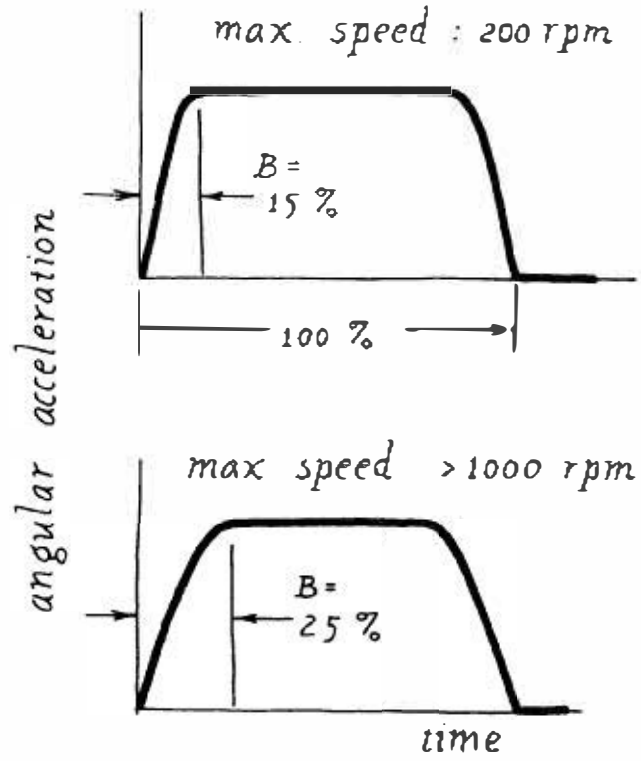


Fig. 8–6. The time required to get from zero to maximum acceleration has an influence on the maximum speed at which a given cam can be driven in these modified trapezoid curves.

The cam designer would also like to keep peak accelerations as low as possible since peak stress in a system is proportional to peak acceleration, and high stress means massive machine members, heavy duty constraints and bearings, and/or rapid wear.

The designer has also traditionally been interested in the rate-of-change of acceleration which is caused by his cam. The greater the rate-of-change, the more nearly he approaches instantaneous change, of course, and the more apt the cam is to produce unwanted vibrations throughout the machine. Cam designers, therefore, have identified a "dimension *B*" shown in Fig. 8–6 on the modified trapezoid curve, as the percentage of total acceleration time that is devoted to the transition from zero to peak

acceleration. Mr. Neklutin, in his excellent book on cam designs (see bibliography in the appendix), says that designers were able to increase the operating speed of modified trapezoid cams from a maximum of 300 rpm to over 1,000 rpm by increasing B from 15 percent to 25 percent of the total acceleration time, as shown in the illustration.

A more recent concern of cam designers has been to minimize the peak input or drive torque requirement for the cam. As explained in Chapter 4, the drive torque curve can differ significantly from the output torque or acceleration curve, and the two cannot usually be minimized simultaneously. In Fig. 8–5, the modified sine curve is more popular than the modified trapezoid, because the former requires a lower peak-drive torque.

The Problem of Crossover

Whatever starts must stop, and thus, every acceleration in a machine is ultimately followed by a deceleration. At some point the driver must become a brake as the acceleration goes from positive to negative. This is called the crossover point in a barrel or roller cam system and is one of the cam designer's problems. Figure 8–7 illustrates a simple barrel cam drive. The input rotates continuously, the output moves intermittently. Figure 8–8 shows what happens at the crossover point in this particular system. When the follower is being accelerated by the driver, the forces on the follower are towards the left, as shown in sketch (a). At the crossover point the follower leaves the right-hand wall of the groove in the drive cam and moves toward the left-hand wall (b). The drive cam then brings the output to rest by applying a force toward the right on the follower, as shown in sketch (c) of Fig. 8–8. The crossover point is defined as that point where a roller follower reverses its direction of rotation about its own axis.

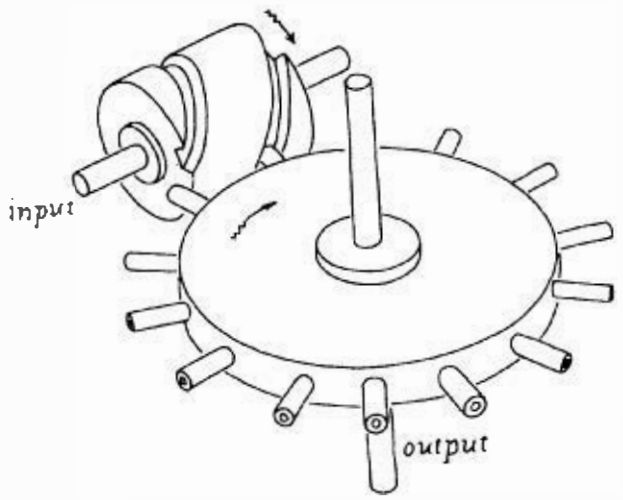


Fig. 8–7. Simple barrel cam—roller gear system.

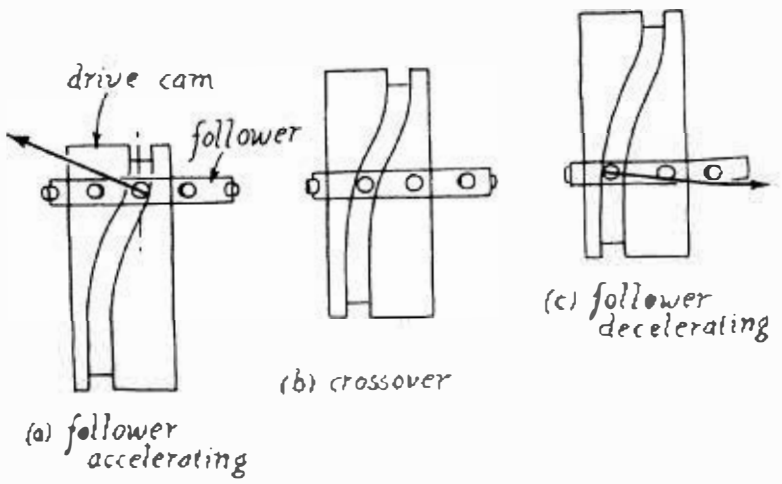


Fig. 8–8. Illustration of the change in forces that occurs on a follower during the crossover point. This is a schematic representation of the intermittent motion cam system shown in Fig. 8–7.

If there is any play between roller and drive cam, then the change from acceleration to deceleration at the crossover point will be sudden and can cause vibration, impact, chatter, etc. There would be play between follower and cam slot in the simple cam system shown in Fig. 8–7 because the follower will inevitably be a roller; in order to roll, it must have some freedom in the slot. More elegant cam systems

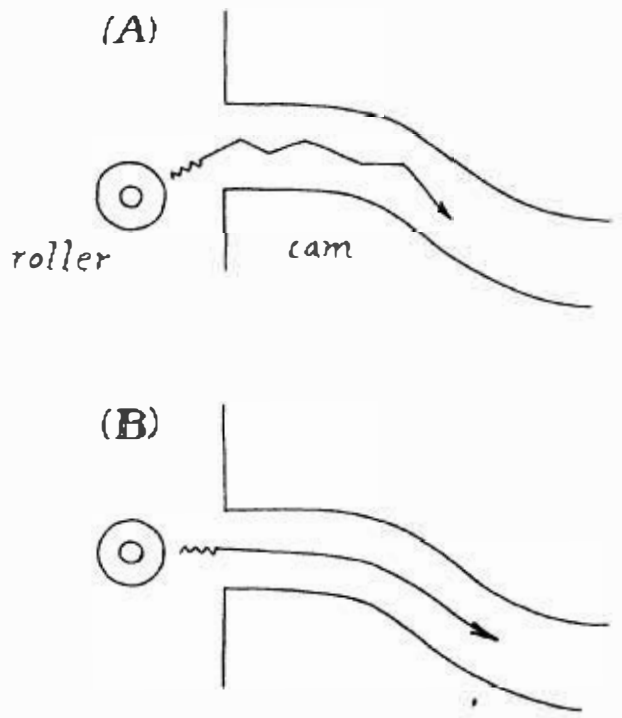


Fig. 8–9. When the follower is driving the cam and enters the drive slot as at A, impact; vibration; chatter; etc.; are liable to occur. These are avoided when the follower enters along the axis of the drive slot, as at B.

use tapered rollers and tapered cam surfaces, as will be seen later, in an attempt to reduce crossover, but it requires very careful machining of the various parts, as well as very careful adjustment of the final system to make much improvement. Nevertheless, the cam man lives in a very precise world and continues to improve it.

Figure 8–9 shows a similar problem with the other class of cam systems in which the follower is driving the cam. Presumably, the cam is at rest when the roller enters the drive slot. If the roller enters at an angle, as shown in (A), it will produce high impact forces, lost motion, bounce, vibration, repeated impacts, chattering, etc. If the roller enters on the axis of the cam slot, as shown in (B), then it can operate more smoothly, depending, of course, on a configuration of the cam slot itself. The designer of Geneva drives must be aware of this problem.

A third problem, similar to the two above, is found in cams of the type shown in Figs. 8–1, 8–22, and 8–23. Here the direction of rotation of the roller followers never changes, so there is no crossover in the strict sense. But there is a point in each motion cycle where acceleration changes to deceleration. At this point one roller picks up the load and another one loses it. Vibration, impact, etc., would again result in a "loose" system.

Figure 8–29 illustrates some of these points, and others. To repeat, a precise cam drive is a superb mechanism, perhaps the best intermittent motion system ever devised; but it must be properly constructed if you are to realize its full potential.

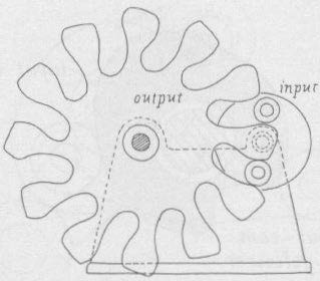
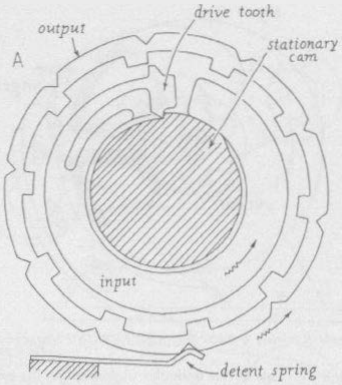
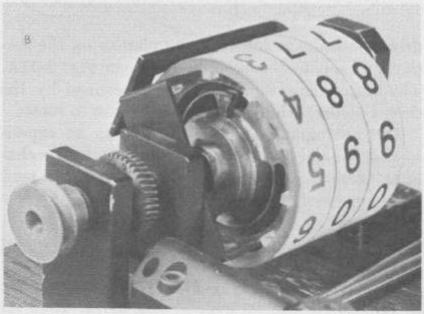


Fig. 8–10. Intermittent motion cam suitable for moderate loads.



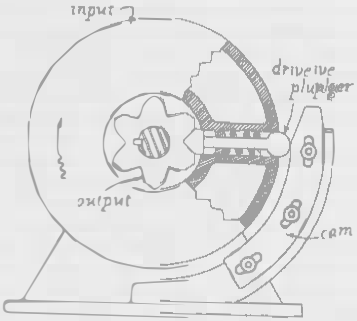


Photograph courtesy of the Veeder-Root Company

Fig. 8-1 [illegible] motion cam suitable only for light loads.

Figure 8–10 shows a simple Geneva-like cam system. Since at least one roller is always engaged with the output cam a locking ring or detent is not required as with the more common [illegible]

The cam system, Fig 8–11, is a stationary shaft. A plastic drive ring is rotated in the direction shown by the [illegible] of the two broken arrows. A tooth mounted on a flexible arm of the [illegible] engages the stationary cam. At about the one o'clock position the drive tooth is moved outward by the stationary cam and picks up one of the square teeth on the [illegible] diameter of the output ring. The output is then moved along with the [illegible] until the



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but heavier loads as long as speeds are modest.

tooth drops down off the plateau on the stationary cam, to the position shown in Fig. 8–11A. The output now remains at rest (retained by the detent) while the input continues to rotate.

The photograph (Fig. 8–11b) shows an experimental counter incorporating this indexing mechanism. However, this type of drive could not be used in a high-speed, high-performance machine, but it is a very low-cost and practical indexing mechanism for small, slow-speed applications where cost is a factor and loads are light.

The device shown in Fig. 8–12 is a heavier example of the plastic cam system of Fig. 8–11. Here all parts are, presumably, metal, and are designed for heavy duty. The input rotates about the output plate and drive plunger to the output wheel at one point during each revolution of the input. By providing additional cam surfaces, of course, the designer could provide more than one dwell per revolution of the input. He would also have a bi of this sort.

The mechanism shown in Fig. 8–13 was invented by Mr. Traugott Tschudi for a valveless, four-cycle, rotary internal-combustion engine. This engine, which was not being mass-produced at the time this book was published, has nevertheless, attracted a good deal of interest. One of its advantages is the fact that it produces fewer air pollutants than does a conventional internal-combustion engine. As a result, the T rated 17 pages in the *States Senate Repor* #90–82. As a fan of intermittent motion devices, I find it very intriguing to think that one of them might contribute materially to a solution to one of the most pressing problems of our times, namely, air pollution. I also find it interesting that intermittent motion applications are important enough to attract the U.S. Senate.

I think it is difficult to understand from a single illus how the T cam system works. I saw a picture of t device it that it would jam. How , it does , as be seen from sequence in Fig. 8–14.

The Tschudi cam system must operate very smoothly to be successful in a high-speed rotary engine. A m the plotted cu Fig. 8–15 teristi

Figure known Ferguson drive, or roller-gear , as igiuated by the on Machine (The ller grar is the output (intermittent motion) The barrel cam, which rotates continuously, supplies Notice m ribs with which it drives the output rollers are tapered. At least two cam at all minimizing crossover effects.

Figures by us size limits Ferguson roller photographs showing these gears to be several feet drives ained accelerat profiles that are modified pezoids, ics, etc.

Fig. 8–15, nd this time, intermitt outputs are driven in synchronism by two cams.

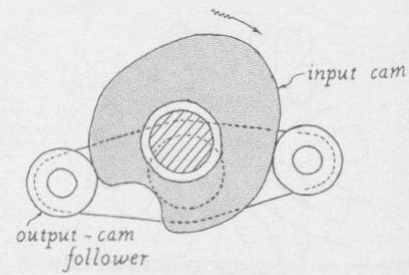


Fig. 8–13. Cam mechanism designed by Mr. T. Tschudi, expressly for a rotary internal combustion engine.

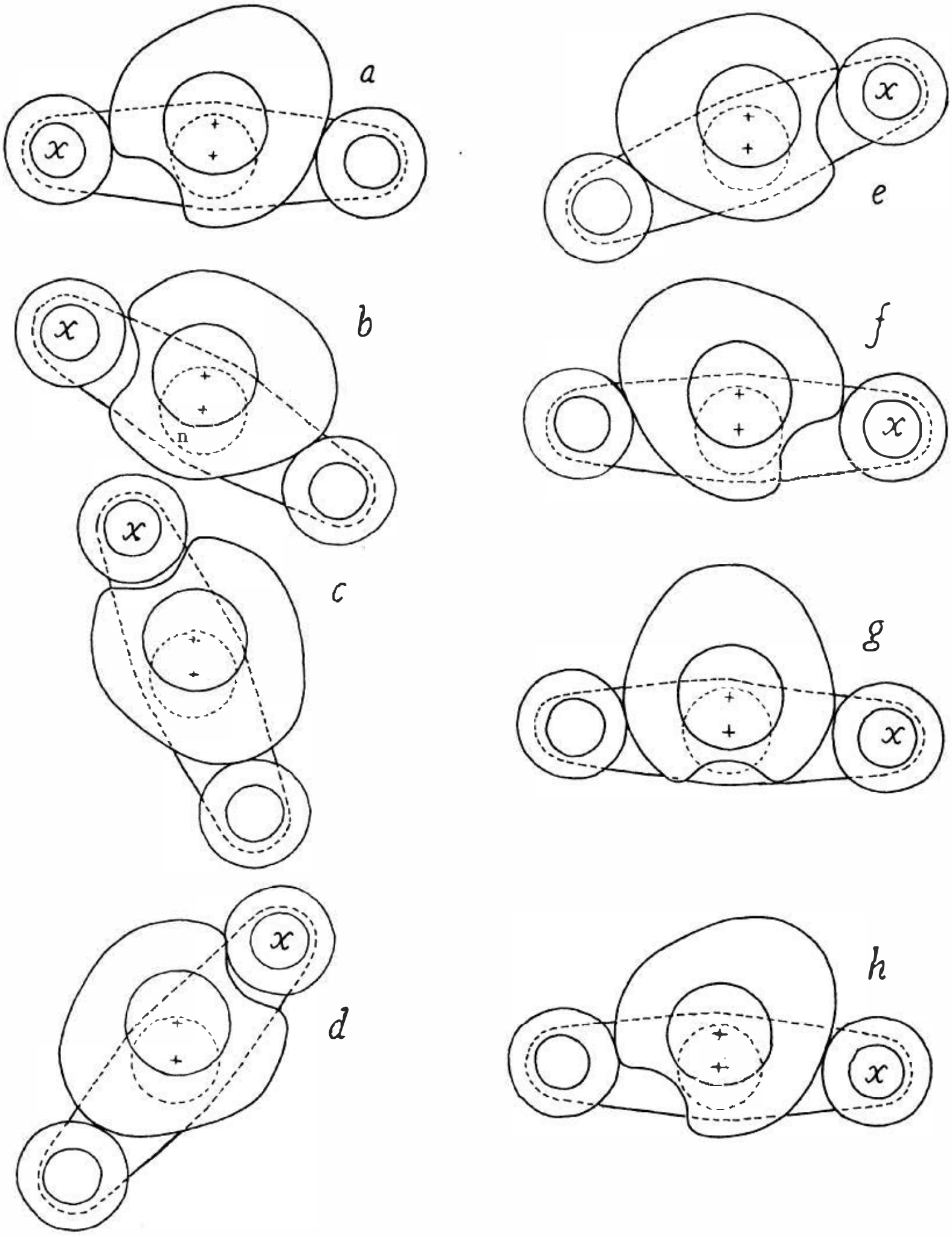


Fig. 8–14. Sequence of operation of the Tschudi cam system.

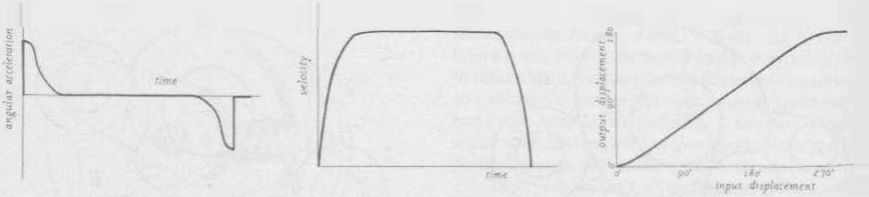
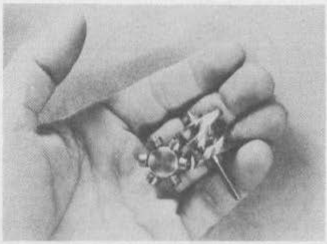
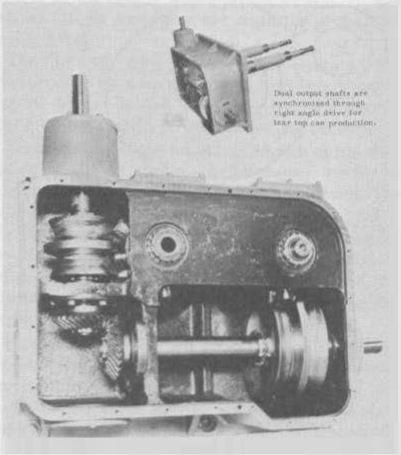


Fig. 8–15. Motion curves for the Tschudi cam system.



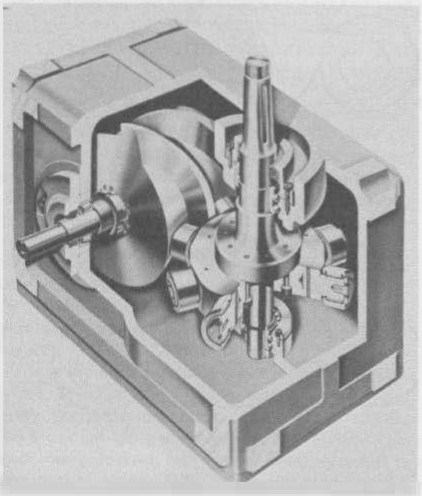
Photograph courtesy of the Ferguson Machine Company

Fig. 8–16. Miniature roller gear indexing drive.



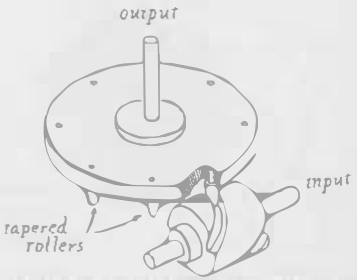
Photograph courtesy of the Ferguson Machine Company

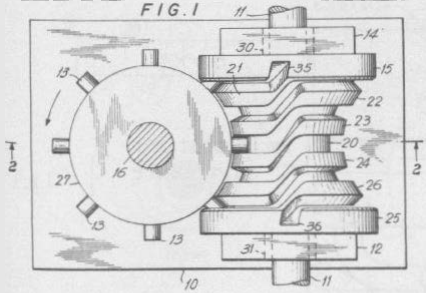
Fig. 8–18. Another Ferguson roller gear system.

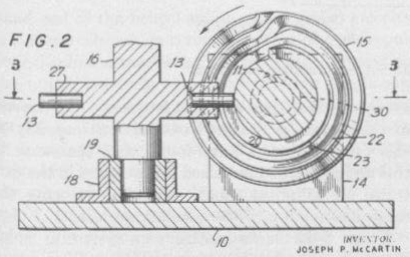


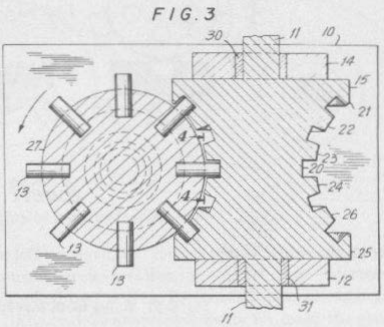
Photograph courtesy of the Ferguson Machine Company

Fig. 8–17. Larger version of the Ferguson drive.







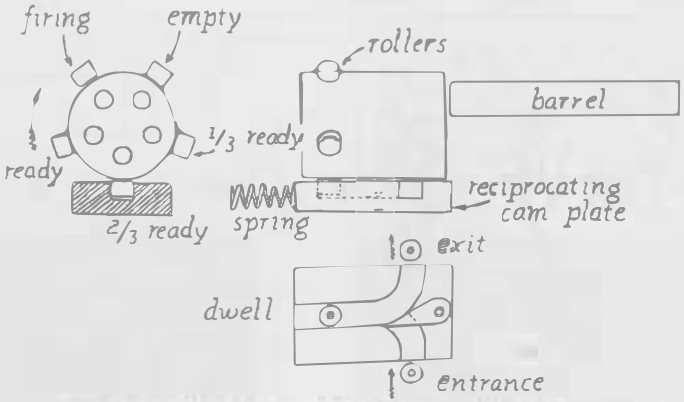


INVENTOR
JOSEPH P. McCARTIN

Figure 8-19 shows a grooved barrel cam indexing system similar to the Ferguson unit in intent, but different in constructional details. Generally speaking, only one follower roller engages the drive cam at one time, although, during dwell periods, two rollers may ride on opposite ends of the barrel cam. Again, tapered rollers and cam surfaces are used to

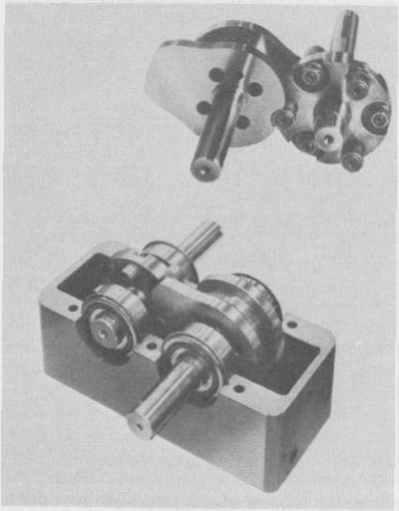
In the barrel-cam arrangement shown in Fig. 8-20 barrel has been modified so that more than one roller on the follower can be engaged simultaneously to the load-bearing capacit of the sy

The next diagram shows a variat on the roller-tooth cam systems seen the last five illustrations. The main difference here is that the input member flat, plate barrel cam. Today's high-speed aircraft cannons work somewhat on the iple of the old C gun a drum full barrels was rotated intermittently past a position. In the cannon sketched Fig 8-21 rum has five positi Looking at the end on the left-hand side, the barrel at the o position empty, while that at the four positi ady to fire. That at ix o'clock two-thirds ready to fire, and finally, the barrel at the eleve 'clock ti is firing. Rotation of the drum is accomplished by a ing face-cam plate, as shown the diagram. The function o am plate, of course, to that of the barrel cam shown Fig 8-19. As the cam plate reciprocat s it engages rollers on the drum and es the drum. Th track at the end of the cam plate dwell period during which the cannon is fired and/or



made ready to fire. A switch plate [illegible] the cam assures that the motion of the drum will [illegible] the

The [illegible] (mass) and [illegible]



Photograph courtesy of the Ferguson Machine Company

Fig. 8-22. Indexing system with coojugate cams.

are operated at their resonant frequency and the rest of the system is designed to function smoothly at [illegible] rate. [illegible] have found that cam wear i[illegible] design can be [illegible]e curves in the cam track are elliptical. Such a track best suits the sinusoidal motion of the oscillating plate

Figure 8-22 shows another cam system in which the output member [illegible] a group of rolle[illegible] is [illegible] concept to that shown in Fig 8-1, and both cam systems can be obtained with the same types of profile (modified sine, modified trapezoid, etc.). Both also are "conjugate" cam systems because two cams, one the mirror [illegible] of the other engage the output at all times maintain perfect control of the output

The version of the conjugate cam system, shown in Fig 8-23, will produce one dwell per revolution of the input shaft, but other versions of this same drive will produce two dwells per input revolution. The preferred acceleration profile for this design is a modified [illegible] curve, but other profiles have been used. Note that the input and output shafts of this device are parallel, unlike many of the roller and [illegible] A, B, [illegible] positions of the cam system.

[illegible] 24 sh[illegible] schematic [illegible] of a [illegible] drive [illegible]n. In the [illegible] a cylindrical drive nut is pushed axially down a helical spline. Th[illegible] velocity c[illegible] rotate the output pinion which is keyed to the right-

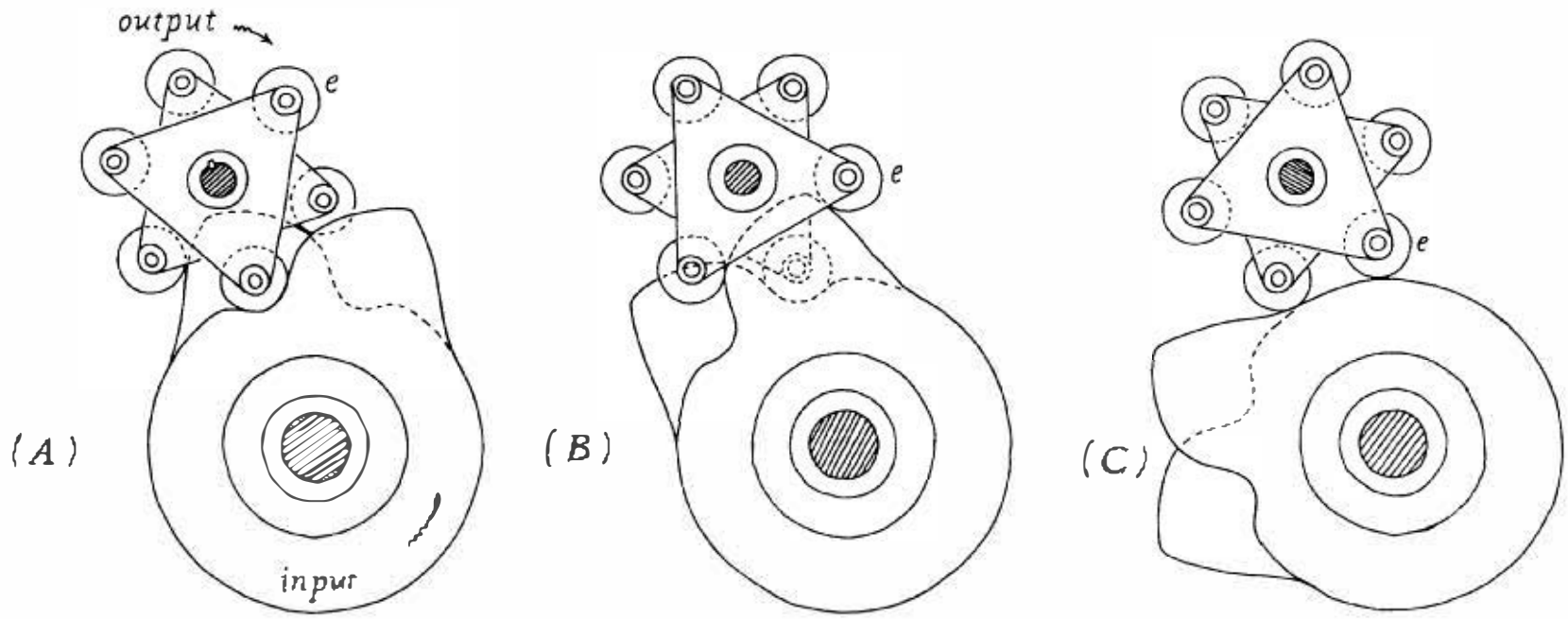


Fig. 8-23. Conjugate cam system with parallel input and output shafts.

hand end of the helical spline. (The pinion velocity is Y rpm.) The action is similar to that of a push-pull helical-spline screwdriver. In the center illustration the nut is not only being pushed down the helical spline, it is also being rotated in a counterclockwise direction. The rotation of the output gear will now be increased to a value $X + Y$; the rotation of the drive nut being added to the rotation produced by the translation of the drive nut down the helical spline. In the bottom illustration, the drive nut is being drawn back along the helical spline. It is also being rotated, still in a counterclockwise direction, while being brought back. Since it is now following the helical spline, there will be no motion in the output pinion during this part of the operation.

If a drive nut of this sort, then, could be continuously rotated in a counterclockwise direction, and at the same time, moved back and forth along the helical spline, the output pinion would index intermittently with a velocity $X + Y$ rpm.

A cutaway view of an indexing drive developed by Roger Gerber, of the New Britain Machine Company, is shown in Fig. 8-25. This drive operates on the principle shown schematically in Fig. 8-24, in which a drive nut (the inside of the cam shown on the right-hand end of the mechanism) is rotated continuously, and is simultaneously reciprocated back and forth along a helical spline to produce intermittent motion in the output gear. In practice, the cam nut is rotated continuously only when the clutch (shown on the left-hand end of the illustration) is engaged and the brake is disengaged. Adding a clutch-brake system to this device allows the machine programmer to provide long dwell periods by electrical control of the input rotation. If the clutch is left engaged, the device will continuously produce output cycles. (Drawing made from information supplied by the New Britain Machine Company.)

Figure 8-26 shows one design for using the general principle illustrated in Fig. 8-24. An input shaft drives a barrel cam and worm gear, both of which are keyed to the input shaft, but are free to slide along it. A roller on the frame engages a slot in the barrel cam and moves the cam gear assembly back and forth as the input turns. Rotation of the worm gear is effectively cancelled or aided by this linear motion and the output worm wheel will move intermittently. This is similar to the motion shown schematically in Fig. 8-24, although, of course, is not identical to it.

A third design for a cam-helical dwell mechanism

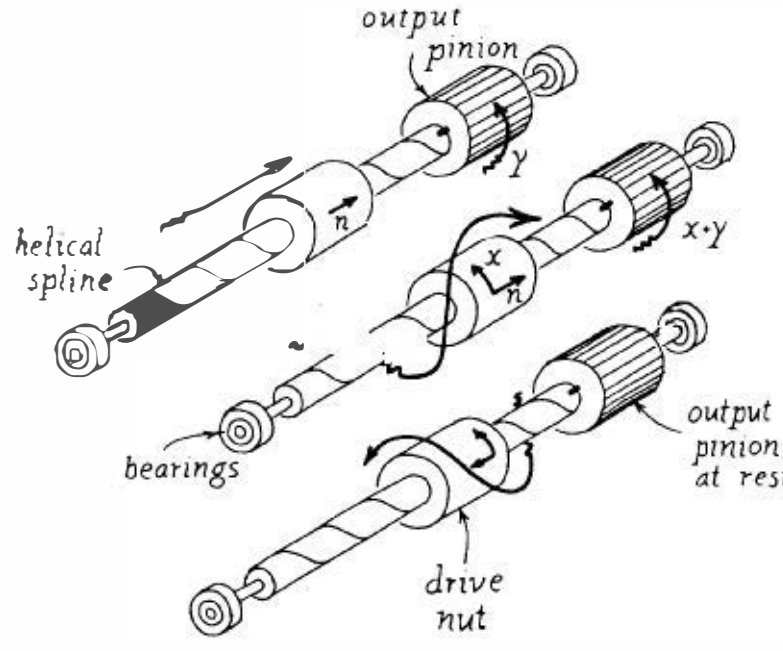


Fig. 8-24. Helical spline-cam system.

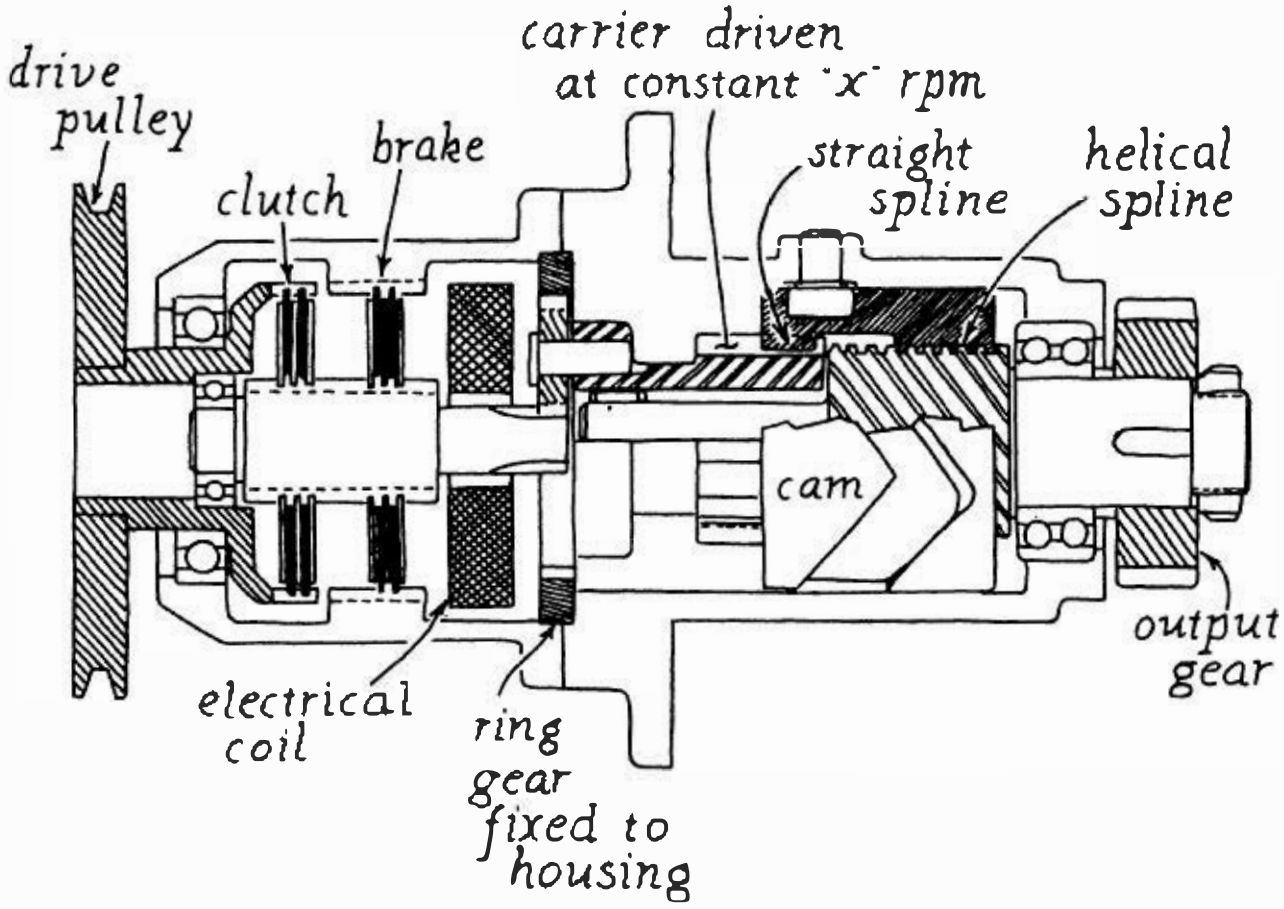


Fig. 8-25. Cutaway view of the indexing drive of a helical spline-cam system.

is shown in Fig. 8-27 and also operates on the general principles shown schematically in Fig. 8-24.

The mechanism of Fig. 8-28 is like an "inverse Geneva" with the cam driving the roller arm follower.

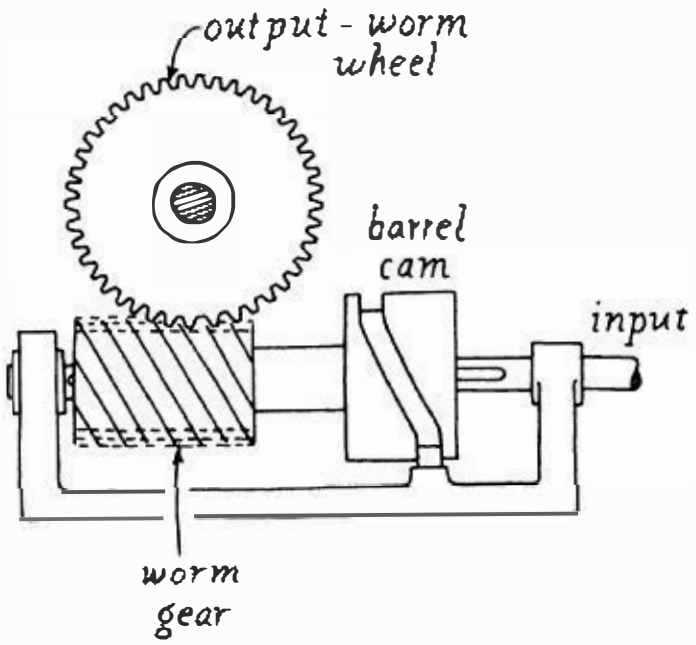


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Fig. 8-26. Another helical spline-cam system.

Since the cam cannot move one roller far enough to bring the next roller into engagement with the next drive slot, the follower must be provided with four leaf springs which engage rollers on the slotted driver. The springs determine the torque this system can deliver to the load.

Anatomy of a Cam

Now let us review some of the main features of a cam system, as shown again in Fig. 8-29, and discuss each component of the system and its function:

A. Follower turret (output) consists of two groups of three precision rollers fastened rigidly together by triangular yolks. The rollers (followers) are yoke mounted, instead of cantilever mounted, for greater ruggedness; B. Two hardened steel cams (input) are shaped alike but are mounted in reverse order, rather like the mirror images of each other. (This can best be seen in Fig. 8-23.) The cams are fastened rigidly together; C. Two rollers (at least) are always in contact with the cams to eliminate backlash and

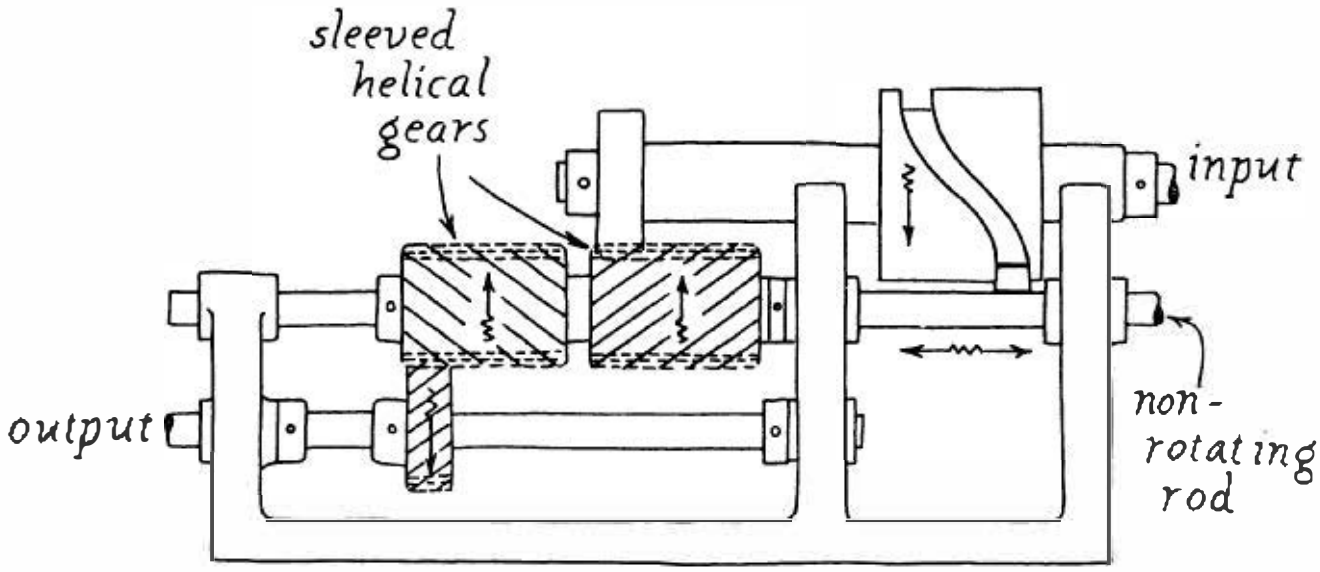

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Fig. 8-27. Cam-helical dwell mechanism.

"crossover" effects. As a result, the follower turret is never free to move independently. Since no springs or other forms of compliance are present, the various elements of the system must be machined with considerable precision to make this possible; D. Input and output "shafts" are parallel and hollow, for convenience. Shafts can be extended on either and/or both sides of the system from either follower turret or cam assembly, for

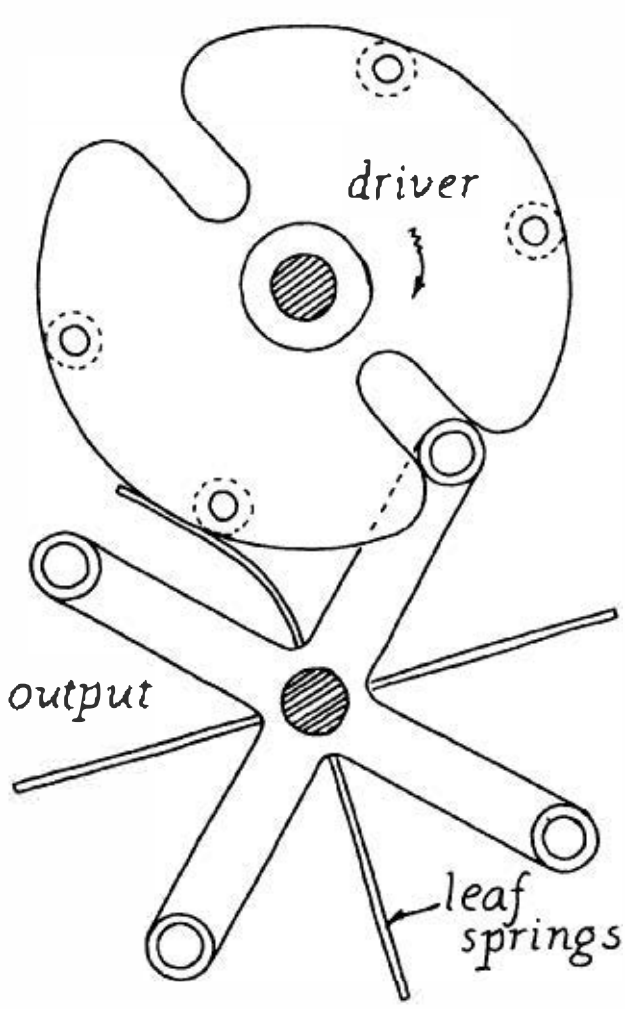
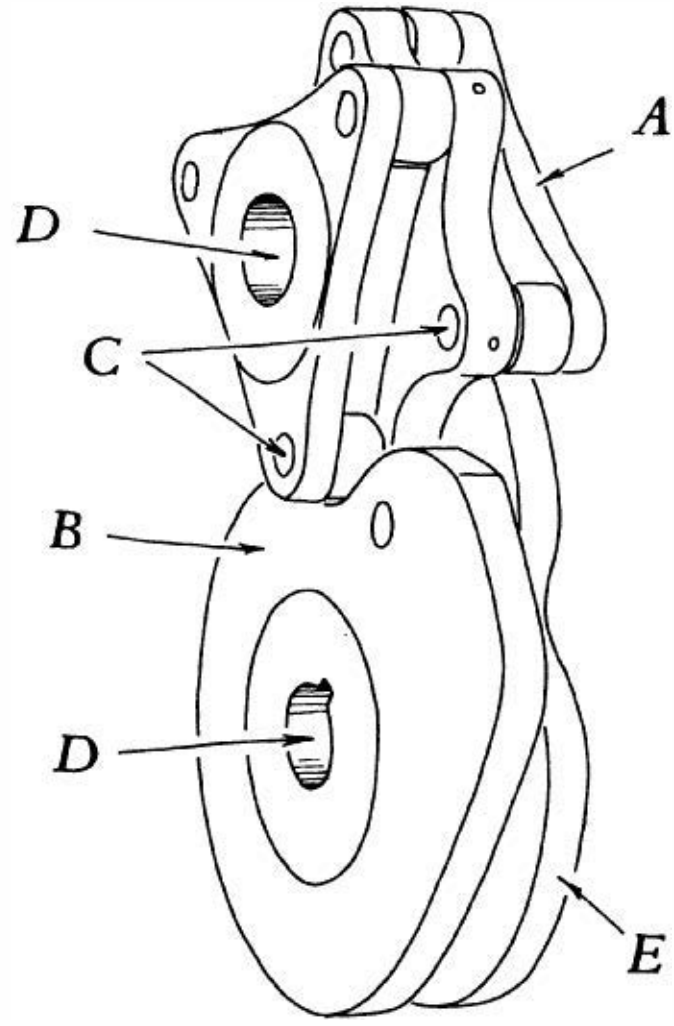


Fig. 8-28. A one-of-a-kind cam system intended for light loads only. (U.S. Patent 3,421,380; M. N. Mansour.)



Drawing courtesy of Commercial Cam and Machine Company

Fig.8-29. The anatomy of a cam.

example. The distance between shafts must be carefully selected and maintained for proper performance; E. The cam profile is a modified sine curve (cycloidic-harmonic-cycloidic) with low peak accelerations, and a relatively long period between peak accelerations. The curve makes a gradual transition to and from zero at the beginning and end of motion, hence, there is no instantaneous change in acceleration (no "jerk"), thereby reducing vibration and shock.

Historical Note

Since man's beginning, new technology has very often found its first application in his weapons. Was the first lever a spear? Or the first ratchet used to wind a catapult or a crossbow? Perhaps not, but certainly these uses were not far behind the original invention. In this chapter we have seen a sophisticated cam system used to index a high-speed aircraft cannon (Fig. 8–21), whose action is similar to that

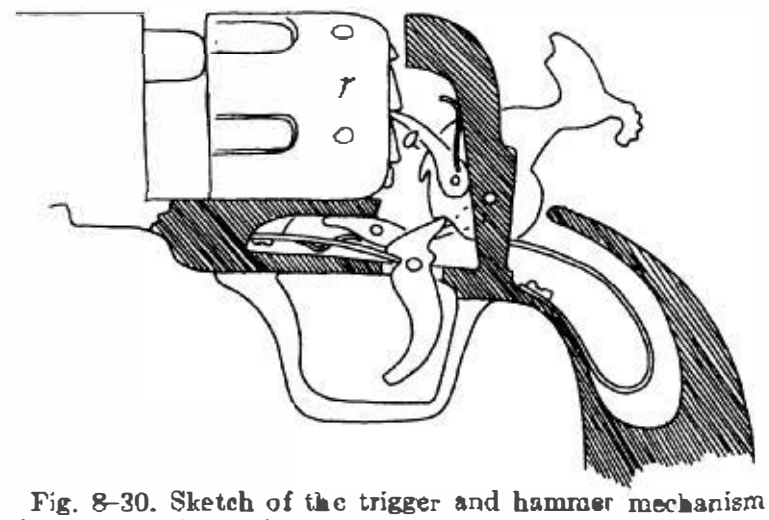


Fig. 8–30. Sketch of the trigger and hammer mechanism of an early Colt revolver.

of the famous (or infamous) Gatling gun. In the illustration, Fig. 8–30, the trigger and hammer mechanisms of an early Colt revolver are shown. Perhaps it is stretching the case a bit to call the trigger a "one-shot escapement," but certainly the chamber (r) is indexed by a ratchet (a).

CHAPTER 9

Geneva Mechanisms

INTRODUCTION

The Geneva is one of the earliest of all intermittent motion mechanisms and when input is in the form of continuous rotation, it is probably still the most commonly used. Genevas are available on an off-the-shelf basis from several manufacturers, in a variety of sizes. They are cheaper than cams or star wheels and have adequate-to-good performance characteristics, depending on load factors and other design requirements. Figure 9–1 shows a typical, four-slot external Geneva.

Advantages of Genevas

Genevas may be the simplest and least expensive of all intermittent motion mechanisms. As mentioned before, they come in a wide variety of sizes, ranging from those used in instruments, to those used in machine tools to index spindle carriers weighing several tons. They have good motion-curve characteristics compared to ratchets, but exhibit more "jerk," or instantaneous change in acceleration, than do better cam systems (the Geneva, you will remember, is a special type of cam system).

The Geneva maintains good control of its load at all times, since it is provided with locking ring surfaces, as shown in Fig. 9–1, to hold the output during dwell periods. In addition, if properly sized to the load, the Geneva generally exhibits very long life. One machine tool manufacturer told me that their Genevas will last 20 years, indexing once every few seconds, on a threeshift basis, driving a spindle carrier weighing one ton. This particular Geneva was about 18 inches in diameter.

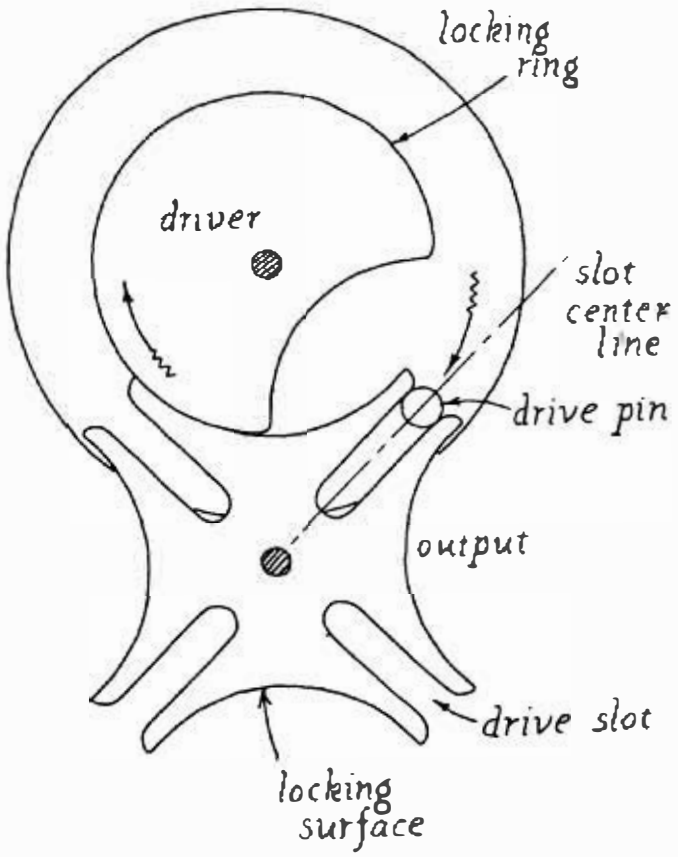
Disadvantages of Genevas

The Geneva is not a versatile mechanism. It can be used to produce no less than three, and usually no more than 18 dwells per revolution of the output shaft. Furthermore, once the number of dwells has been selected, the designer is well locked into a given set of motion curves. The ratio of dwell period to motion period is also established once the number of dwells per revolution has been selected. Also, all Geneva acceleration curves start and end with finite acceleration and deceleration. This means they produce jerk.

Types of Genevas

There are three types of Genevas: (1) external, which is the most popular, and which is represented by the device shown in Fig. 9–1; (2) internal, which is also very common and is illustrated by Fig. 9–2; and (3) spherical, Fig. 9–3, which is extremely rare. We will take a closer look at some external and internal Genevas later on, but will not spend more time on the spherical type.

Genevas are also combined with a wide variety of other mechanisms, such as four-bar linkages; clutch-brake combinations; noncircular gears; etc., to modify the motion curves and dwell-motion ratios obtained from a pure Geneva. We will see several different examples of these later.



Drawing courtesy of MACHINE DESIGN Magazine: Dec 23, 1965: p. 121 ff

Fig 9-1. Fourslot external Geneva.

Motion Curves

The motion curves for several external Genevas are shown in Fig. 9-4; and for internal Genevas, in Fig. 9-5. As can be seen, the two curve sets differ quite drastically in shape. Figure 9-6 shows the motion curves for fourslot internal and external Genevas superimposed for comparison.

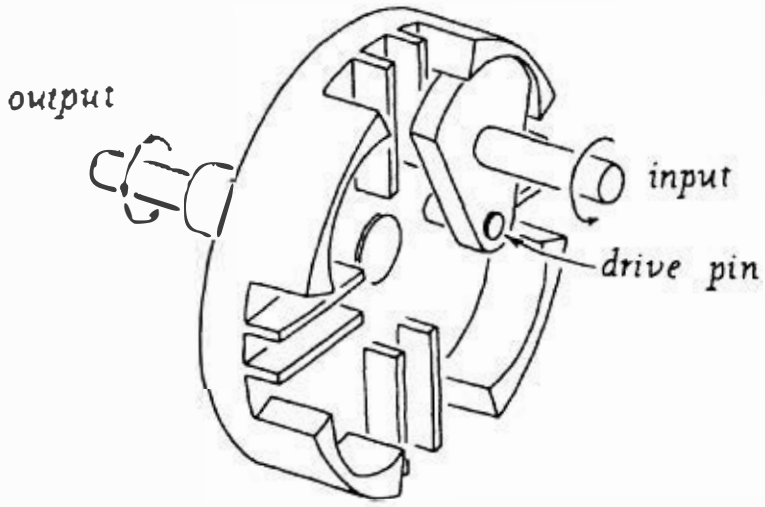


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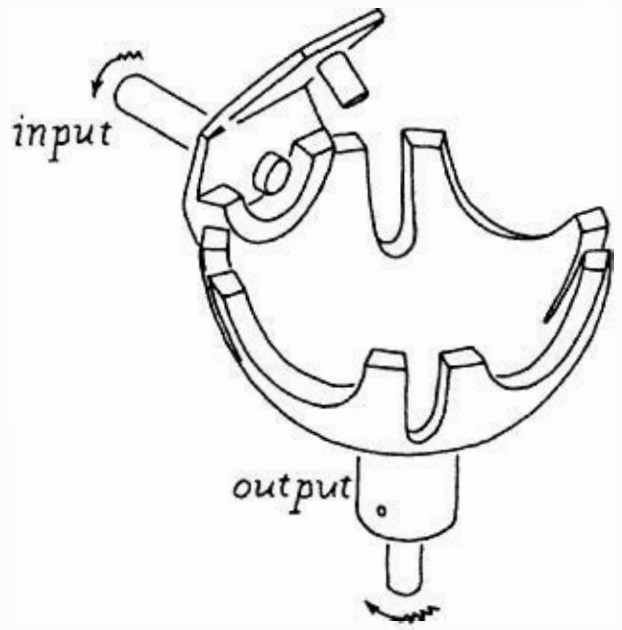
Fig. 9-2. Fourslot internal Geneva.

A few general comments can be made about these motion curves.

1. For an external Geneva, the dwell period always exceeds the motion period.
2. For an external Geneva, the dwell period always lasts longer than the time required for 180 degrees of motion of the input driver.
3. For an internal Geneva, the motion period always exceeds the dwell period.
4. For an internal Geneva, the dwell period is always shorter than the time required for 180 degrees of input.



Drawing courtesy of MACHINE DESIGN Magazine: Dec. 23, 1965: p. 121 ff

Fig. 9-3. Four-slot spherical Geneva.

5. For a spherical geneva, the dwell period equals the motion period, and equals the time required for exactly 180 degrees of input.
6. The magnitude of peak acceleration and deceleration, velocity, etc., obtained with a Geneva, is a function of the number of slots or dwells. This is true of all types.

As the ratio between the diameter of the wheel and the diameter of the driver gets larger (more slots or dwell periods per revolution of the output), maximum accelerations and velocities decrease (for a given driver speed). This makes sense because the indexing angle of the output decreases as the number of slots increases.

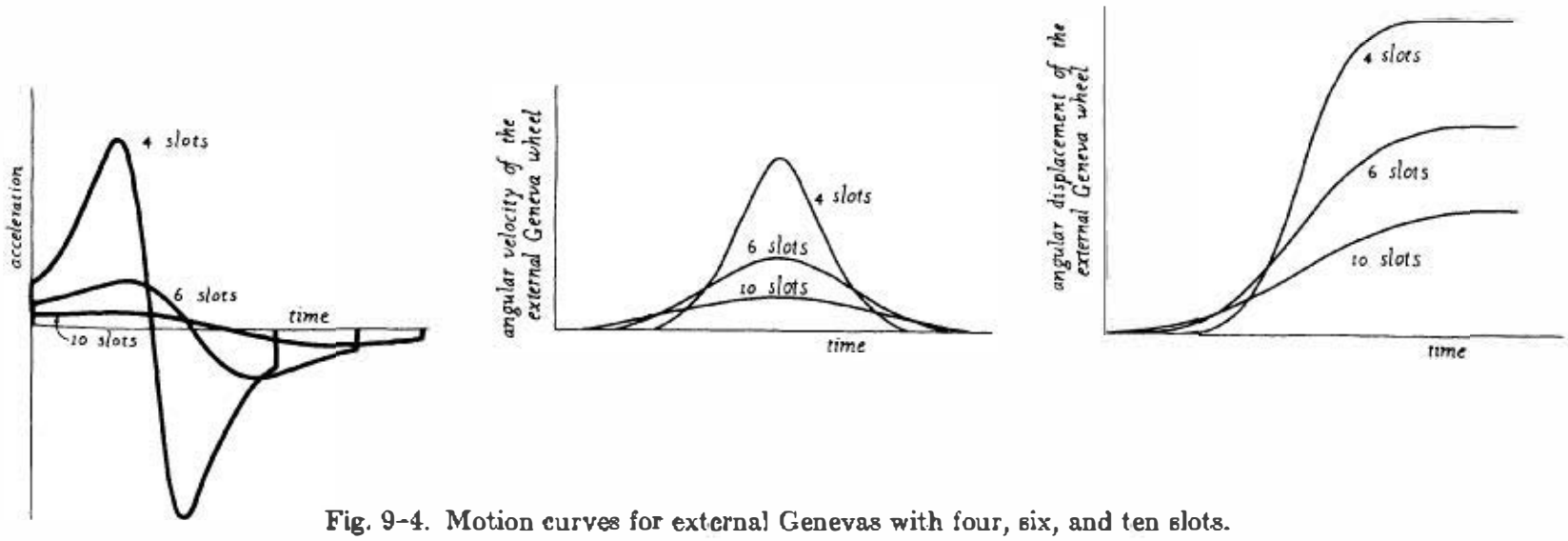


Fig. 9-4. Motion curves for external Genevas with four, six, and ten slots.

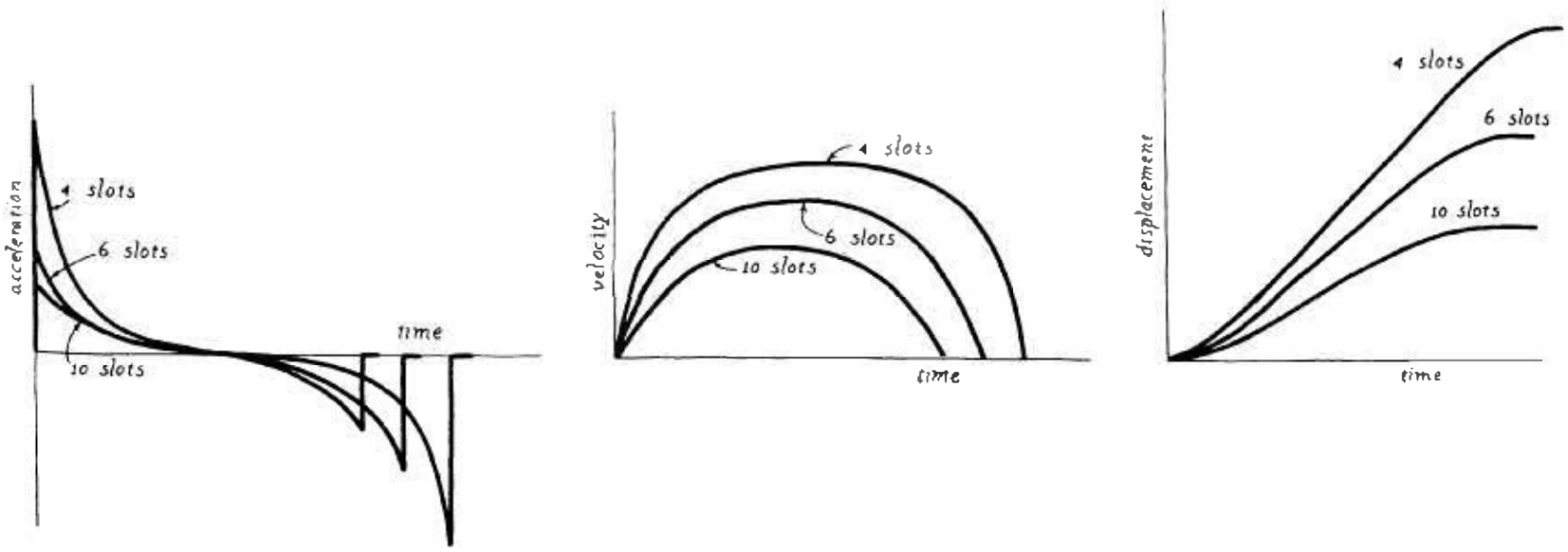


Fig. 9-5. Motion curves for internal Genevas with four, six, and ten slots.

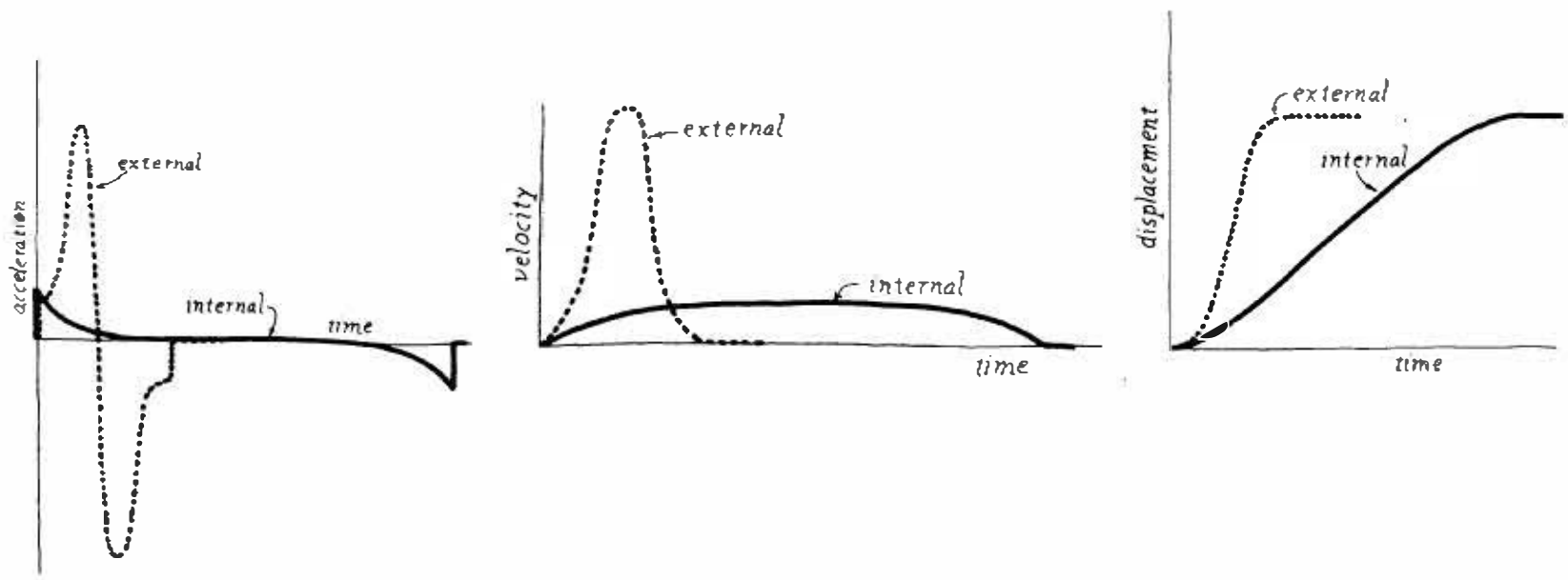


Fig. 9-6. Comparison of the motion curves of internal and external Genevas with four slots.

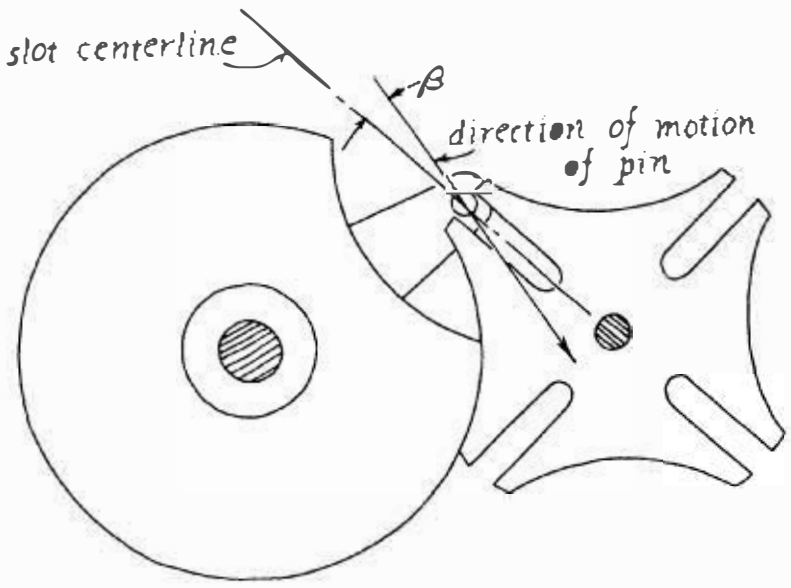


Fig. 9–7. The drive pin of the Geneva should enter the slots in the driven wheel along an arc that is tangent to the centerline of the slot, rather than along a line which deviates from the slot centerline by some angle β, as shown here.

Miscellaneous Comments

In a well-designed Geneva the drive pin will enter the slots of the output wheel along an arc that is tangent to the centerline of the slot, as shown in Fig. 9–1. If the direction of motion of the pin deviates from the slot centerline by an angle β, shown in Fig. 9–7, impact can result. This will either be a positive or negative impact (that is, it will produce either a positive or a negative torque on the driver) depending upon whether the slot is at some angle above the tangent line of the pin's motion or at some angle below it. (Figure 9–7 shows a situation where the tangent line is below the pin's motion line.)

Figure 9–8 shows the displacement curves that result from three situations: the deviation (angle β) is ± 5 degrees or 0 degrees (tangent). There does not seem to be much difference between these three curves, each of which represents only the first few degrees of motion of the output member rather than the complete displacement curve, as per Fig. 9–4. The velocity curves for these three situations, also shown in Fig. 9–8, do not appear to have changed much either.

If we now plot the acceleration curves, however, we will find that we have introduced impact when the slot's centerline was above or below the line tangent to the pin's motion. This is because the angular velocity in each case starts at a finite positive or negative value; and instantaneous change in velocity means an infinite acceleration which, in turn, means impact. In practice, of course, a small initial velocity would mean a relatively small impact (even though theory says any instantaneous change means an infinite acceleration) and would cause no trouble. If the machine is operated at high speeds, however, or under heavy loads, or with elastic connections between parts, even a small, sudden change in velocity can cause severe impact, stress, vibration, lost motion, jamming, etc. Figure 9–9 shows the acceleration curves for the three situations.

As discussed in Chapter 5, a high-speed or heavily loaded Geneva can also get into difficulty if the pin is not a good running fit in the slots in the output

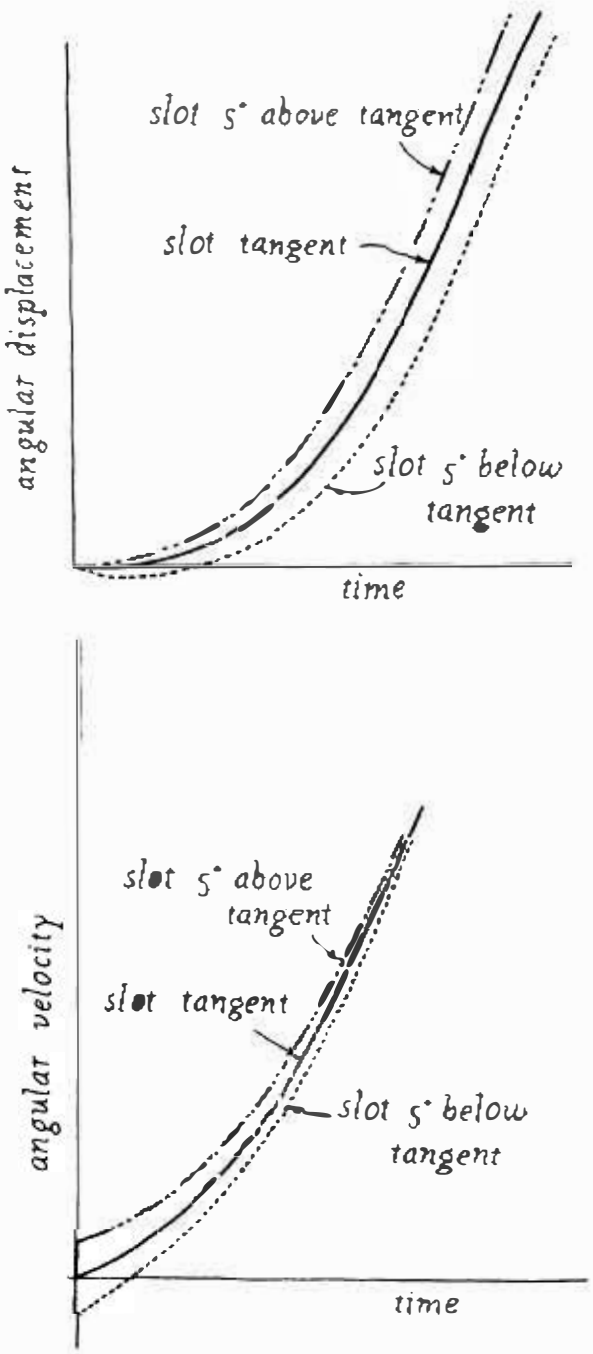


Fig. 9–8. Displacement and velocity curves of an external Geneva when the direction of motion of the drive pin is tangent to the slot; when the slot is 5 degrees above the tangent line; and when it is 5 degrees below it (the latter situation is shown in Fig. 9–7).

wheel. Elasticity in the system may allow the pin to chatter from one side of the slot to the other as it drives the wheel, producing a series of impacts which can cause all sorts of trouble. (See Figs. 5–9 and 5–10 for the resultant motion curves.)

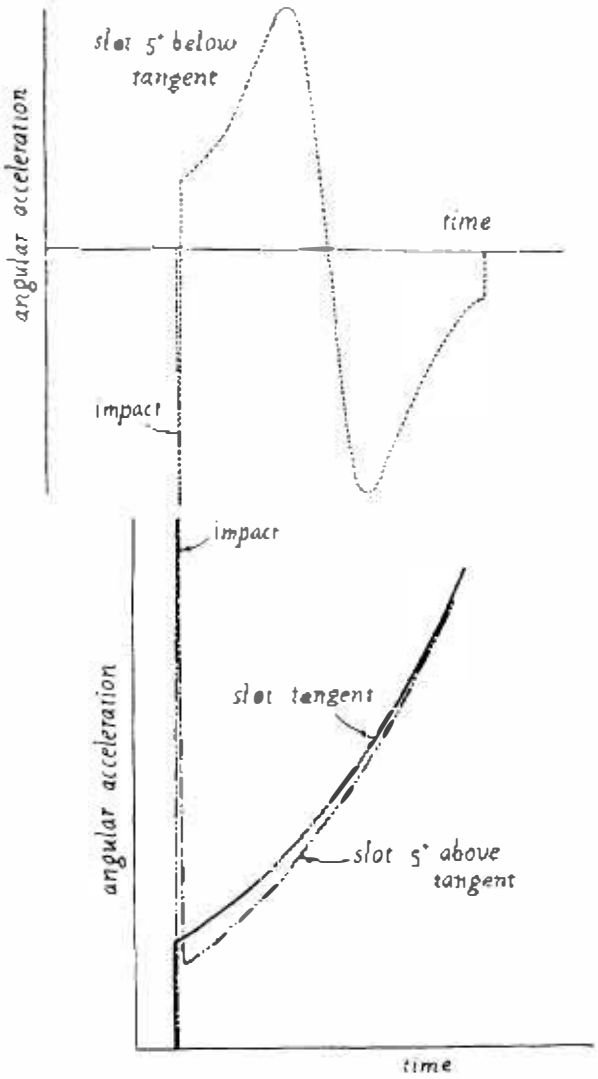


Fig. 9–9. Acceleration curves for tangent and ±5 degrees misaligned slots.

Four sketches of external Genevas with five, six, eight, and ten slots, respectively, are shown in Fig. 9–10. Notice how the ratio between the diameter of the driver and the diameter of the wheel decreases as the number of slots increases. Notice also, that the indexing angle decreases.

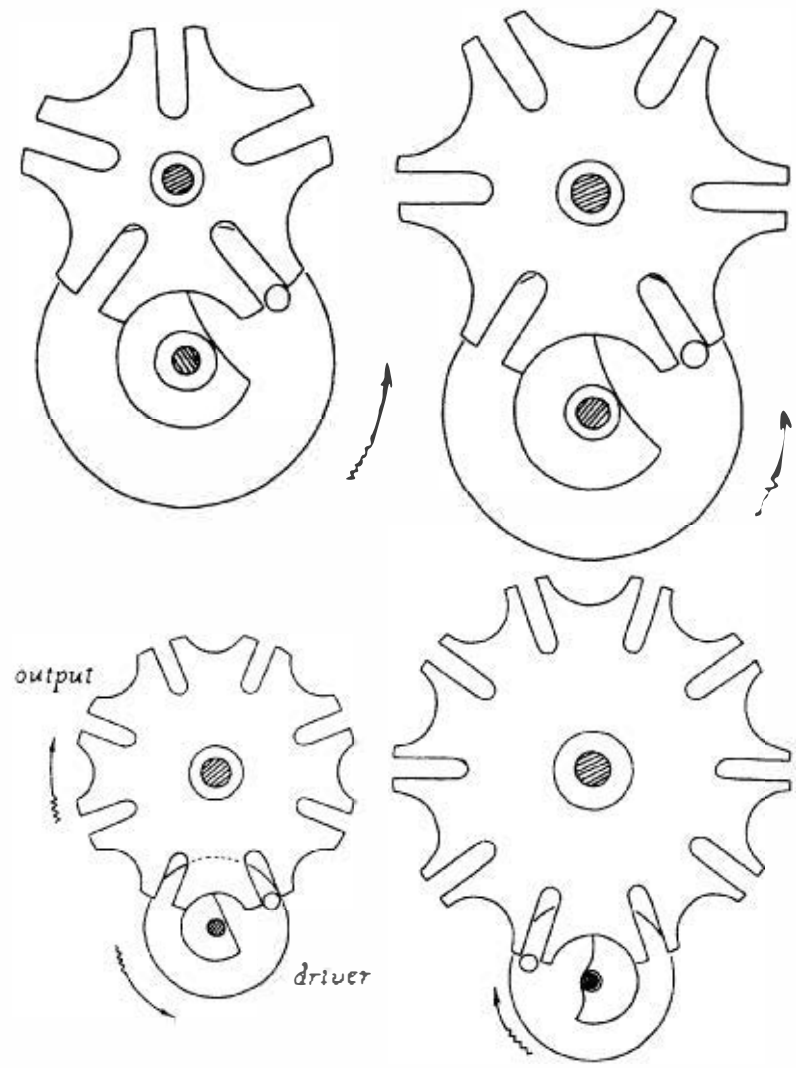


Fig. 9–10. External Genevas with five, six, eight, and ten slots, respectively.

Three internal Genevas with four, eight, and twelve slots are illustrated in Fig. 9–11. Again, notice how the ratio of driver diameter to output wheel-diameter decreases as the number of slots increases. Notice too, that the driver must be mounted on a cantilevered shaft in each case, whereas the external Geneva of the previous illustration could have both input and output members mounted on through

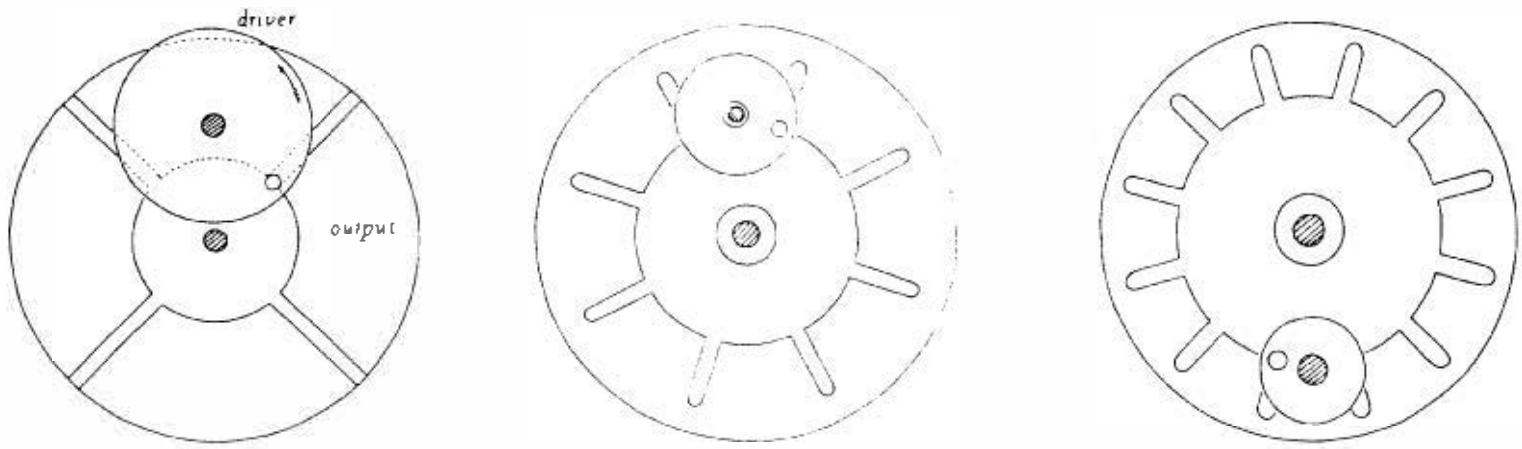
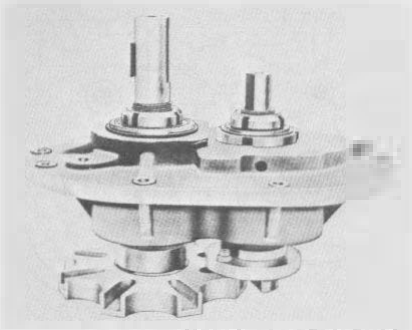


Fig. 9–11. Internal Genevas with four, eight, and twelve slots.



shafts. Some designers [illegible] disadvantage [illegible]

Figure 9–12 shows a packaged [illegible] Four, five, six, or eight slot designs are available in several sizes. Various manufacturers provide ready-made [illegible]

In Fig 9-13, a Geneva [illegible] can be seen between the first and second wheels of a small instru- [illegible] tee impacts [illegible] d otherwise be obtained [illegible] use of mutilated pinions (described in Chapter 10) in the machine. The counter is about three inches long.

The drawings of Fig. 9-14 show a Geneva mechanism used in the indexing system of a large multiple-spindle machine tool. This mechanism is a five-step [illegible] The drive slots [illegible] adjustable members to allow perfect alignment between the slot centerline and the direction of motion of the drive pin as it enters a slot. A Geneva is used for rough indexing of the spindle, while fine indexing is accomplished by cam-driven locating fingers that are not shown in this illustration. (See Fig. 1–1.)

A six-slot external Geneva mechanism (Fig. 9–15) is used for light-duty instrument applications. The configuration of both driving and driven members has been altered to provide a longer locking surface [illegible] driven wh[illegible] could be [illegible] with a conventional six [illegible] Geneva. In other [illegible] higher [illegible] one result of the changes, [illegible]ver, the drive [illegible] must now enter the [illegible] driven wheel at [illegible] angle, rather than enter them tangent to the slot [illegible] ave seen, this will cause impact, etc.; therefore, this design could only be used at slow speeds or with light loads. Furthermore, the entrance to the slot has been chamfered, so that the drive pin will tend to align the output member as it enters the slot, if there has been any preliminary motion or misregistration in the driven member.

Output to input motion ratios can be altered by providing multi-pin drivers, as with the four-slot external Geneva with a three-pin driver, shown in Fig. 9–16. Although the designer will find that he does not have a great deal of flexibility—two, three, and sometimes even four pins are possible.

Figure 9–17 shows a four-slot Geneva driven by a two-pin driver. In the arrangement shown, the output will dwell for a long period and a short period, alternately, as the input rotates.

Very long dwell periods can be obtained if the drive pin is mounted on a chain, as shown in Fig. 9–18, instead of on a wheel. One of the pins in the chain is replaced by an extra-long pin which serves as a driver. More than one chain pin could, of course, be replaced by a drive pin. Most links in the chain [illegible] provided [illegible] locking surfaces for the Geneva during dwell periods. [illegible] ew links [illegible]

The external Geneva [illegible] [illegible]stration, Fig. 9–19, [illegible] by [illegible] disc. This accomplishes several things. The dwell period of the Geneva is increased with respect to [illegible] [illegible]tial and fi[illegible] [illegible]celerations

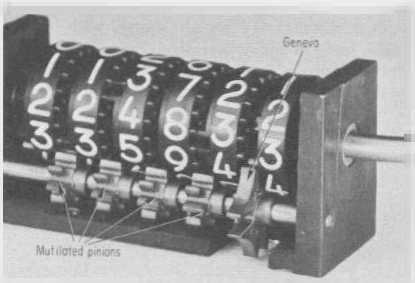


FIG. 1

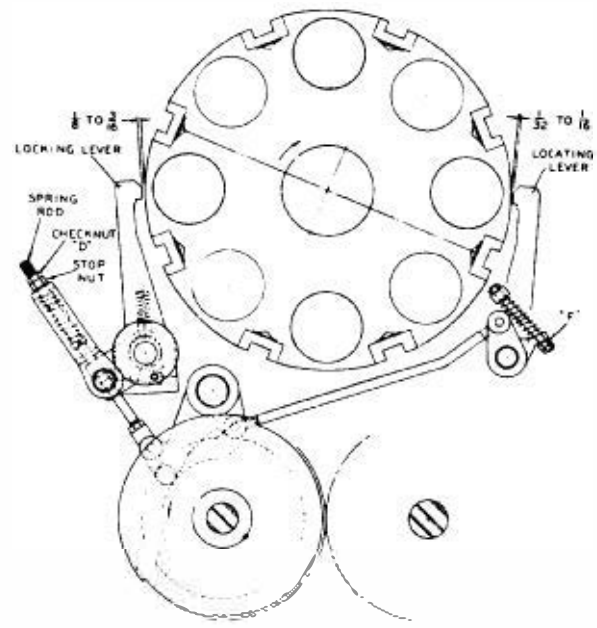
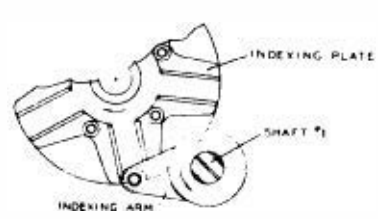
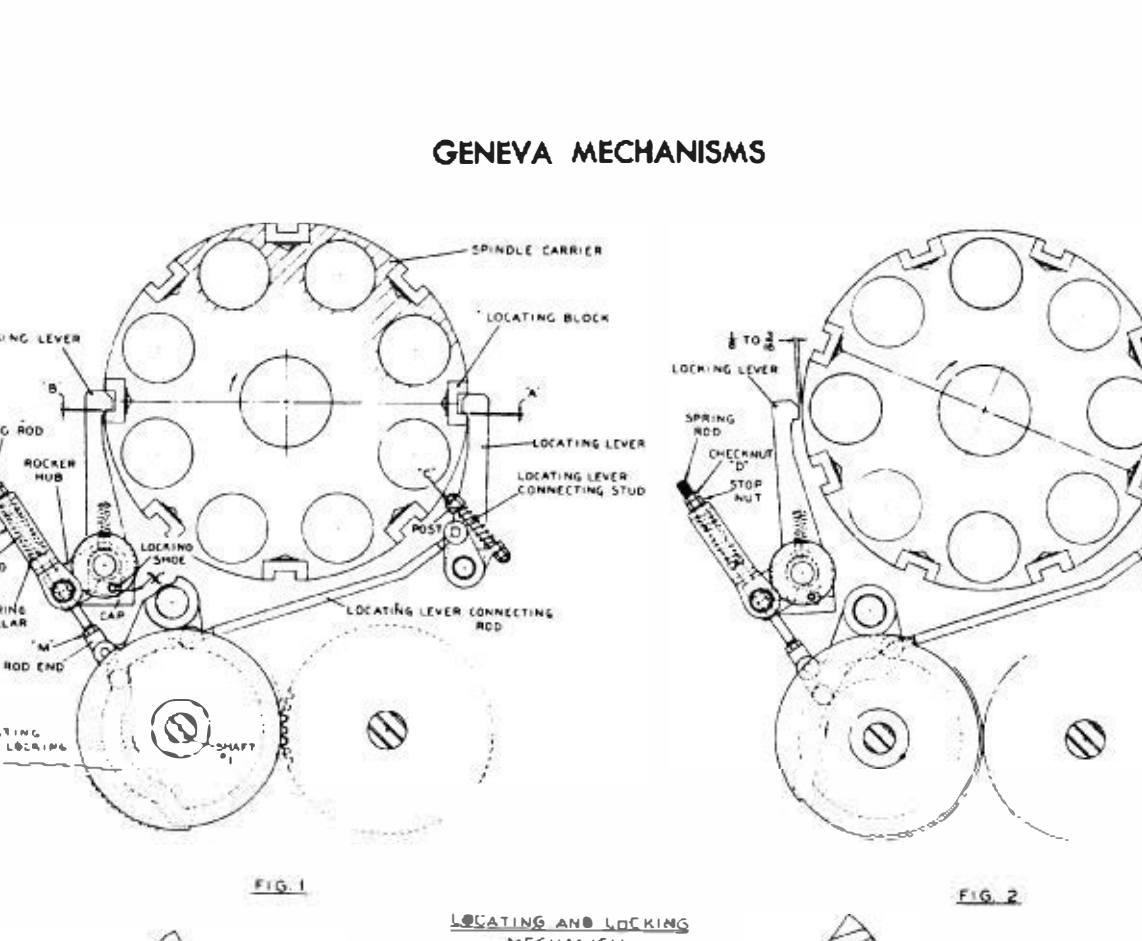


FIG. 2

LOCATING AND LOCKING MECHANISM





CARRIER LIFTING AND INDEXING MECHANISM

Drawing courtesy of the New Britain Machine Company

Fig. 9–14. Five-step Geneva mechanism used in the indexing system of a large multiple-spindle machine tool.

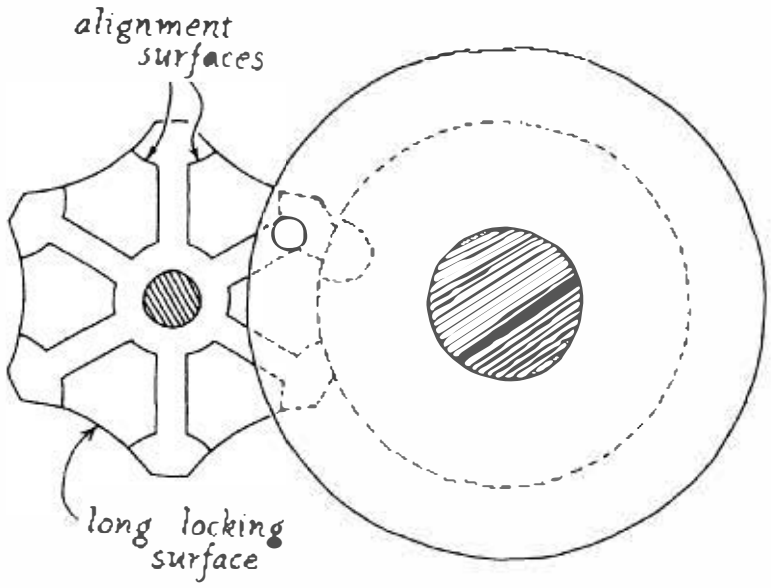
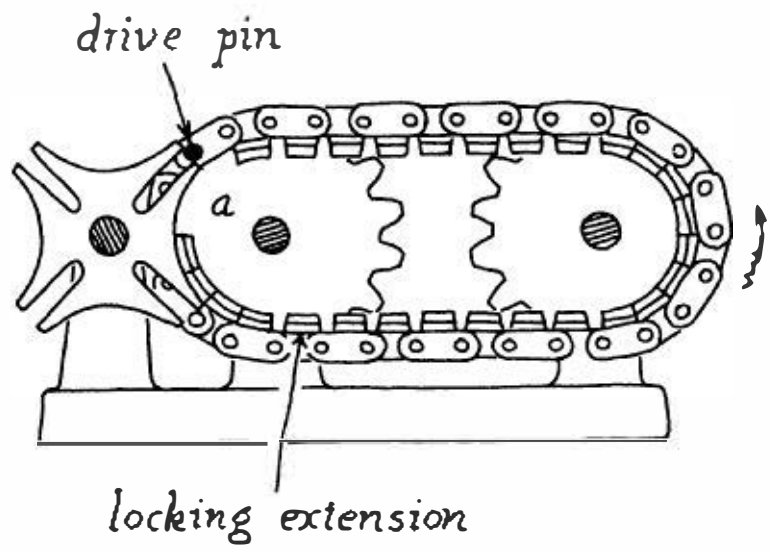


Fig. 9-15. Six-slot external Geneva used for light-duty instrument applications.



Drawing courtesy of PRODUCT ENGINEERING Magazine; June 8, 1964: pp. 67, 68

Fig. 9-18. Chain-mounted drive pins with blocks for locking during dwells.

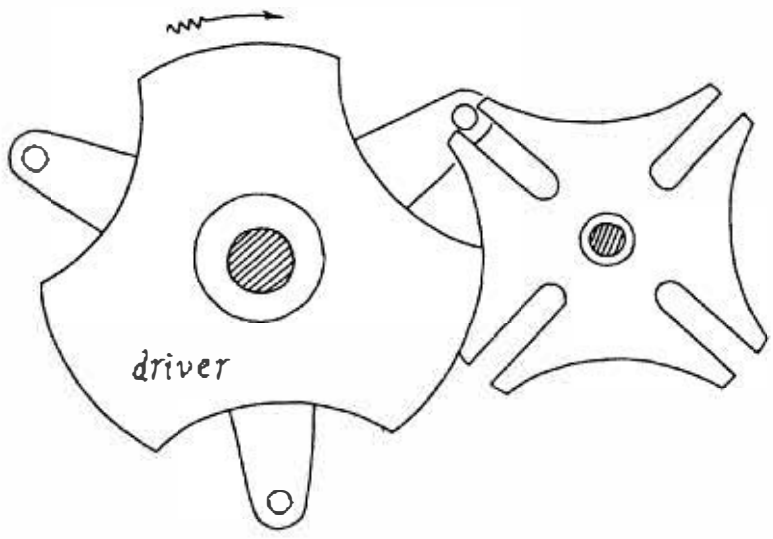
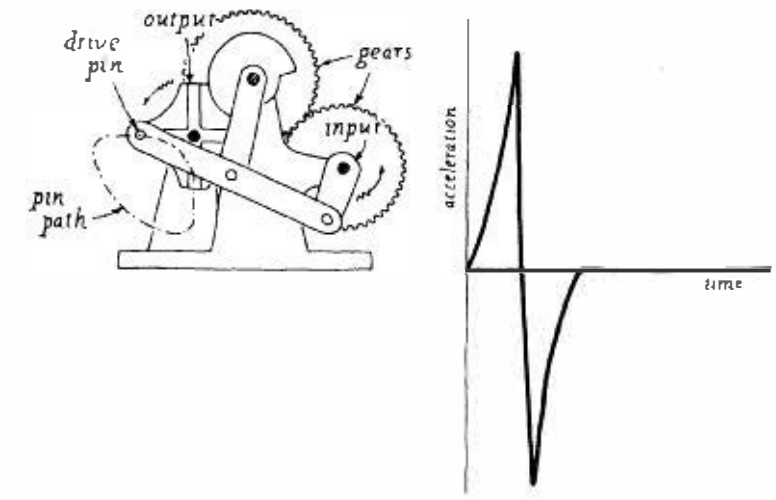
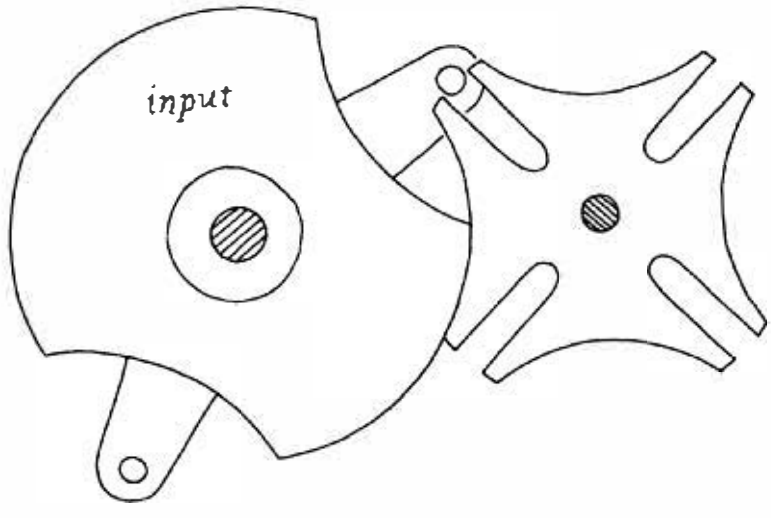


Fig. 9-16. Four-slot external Geneva with threepin driver.



Drawing courtesy of PRODUCT ENGINEERING Magazine; June 8, 1964: pp. 67, 68

Fig. 9-19. (*Left*) External Geneva driven by a four-bar linkage. (*Right*) Acceleration curve.



Drawing courtesy of PRODUCT ENGINEERING DESIGN MANUAL, 1959: pp. 254, 260

Fig. 9-17. Four-slot Geneva driven by a two-pin driver.

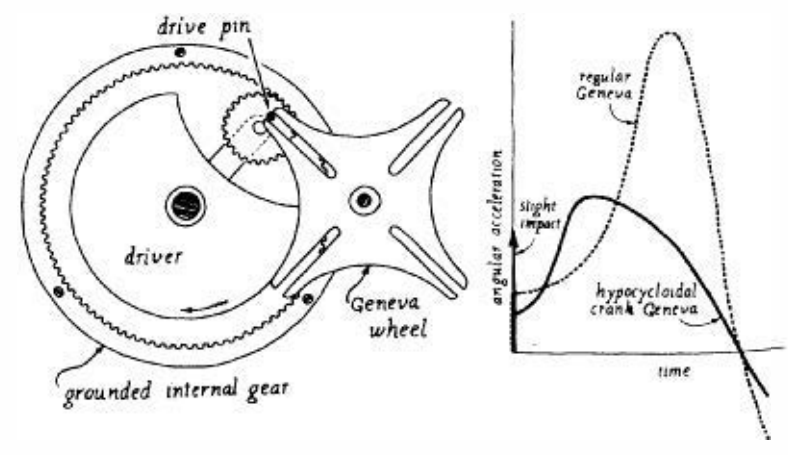
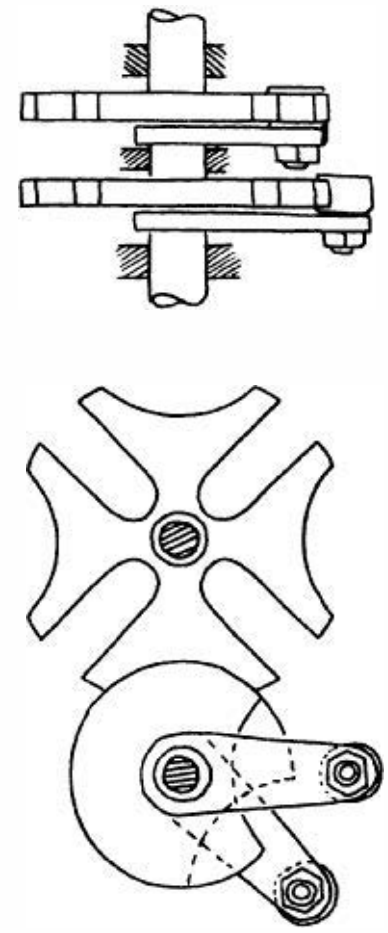


Fig. 9-20. (*Left*) Fourslot external Geneva driven by a hypocycloidal crank mechanism. (*Right*) Comparative acceleration curves.

are zero, not finite, thus eliminating the jerk associated with a conventional Geneva. Since the four-slot Geneva wheel must still rotate 90 degrees with each drive cycle, however, and is now doing it in a shorter length of time (longer dwell period), peak accelerations and velocities are greater here than for a conventional Geneva, as suggested by the acceleration curve.

A four-slot external Geneva driven by a hypocycloidal crank mechanism is illustrated in Fig. 9–20. A gear train is used to reduce the velocity of the drive pin at the moment it enters the slots in the Geneva wheel. The acceleration curves provided here show that there is a "theoretical" slight impact as the drive pin starts the Geneva in such a situation, but in actual practice this impact probably would not be noticeable. A hypocycloidal drive also reduces peak acceleration as can be seen by comparison with the acceleration curve for a regular four-slot Geneva.



Drawing courtesy of PRODUCT ENGINEERING Magazine; Oct. 26, 1964; pp. 109, 110

Fig. 9–22. Twin Geneva drive. Note in the top view that the two Geneva wheels are in exactly the same position.

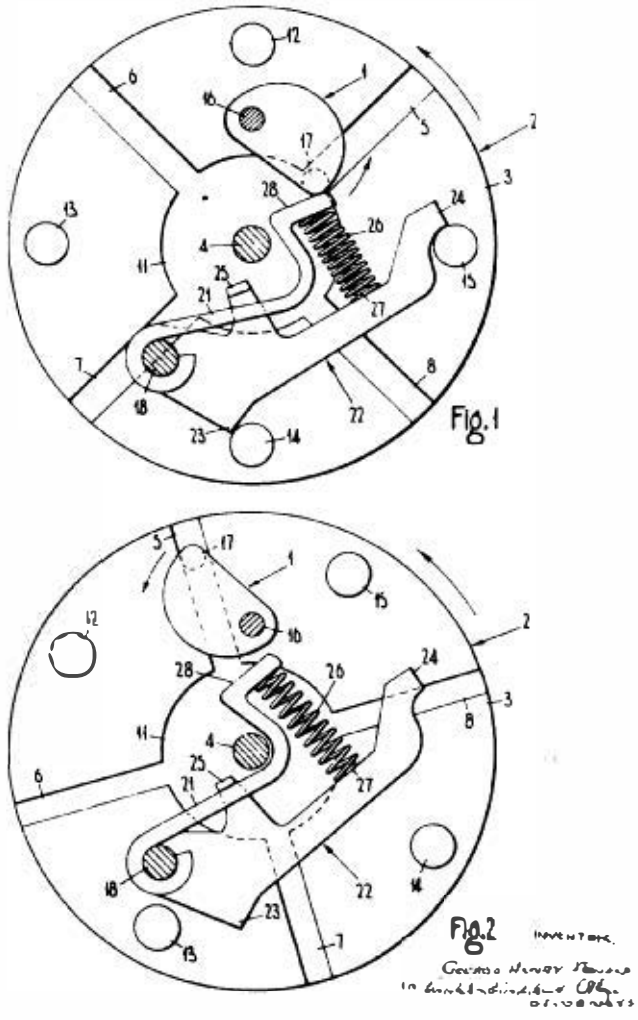


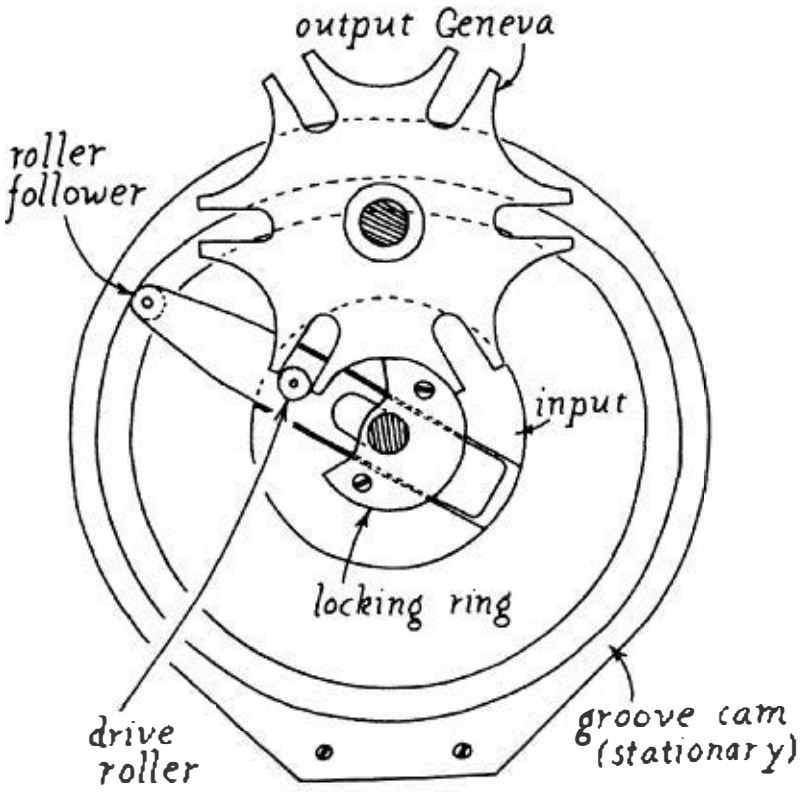
Fig. 9–21. Fourslot internal Geneva with friction grip control of output. (U.S. Patent 3,413,874; G. H. Arnold.)

In the illustration here, only half of the acceleration curve is shown for each device.

A friction grip is sometimes used instead of a locking ring to control the output member during dwell periods. In the diagram (Fig. 9–21) of a four-slot internal Geneva, the locking member *28*, engages the output shaft *4*, to which the output wheel *2*, is connected. Cam *1*, containing drive pin *17*, regulates the motion of the locking member. This friction lock would probably only be satisfactory for light-duty applications.

Figure 9–22 illustrates a twin Geneva drive. The output from one Geneva can act as a driver, or input, to a second. Gears can also be introduced between the two, or on the output of the second to modify the motion even further. Tricks of this kind allow the designer to extend dwell periods, etc.

The drive pin in the groove-cam Geneva design shown in Fig. 9–23 does not operate on a fixed radius of the input member. Instead, its radius (and hence its tangential velocity) is controlled by a groove cam which moves the drive pin radially inward or outward, at the option of the designer, to



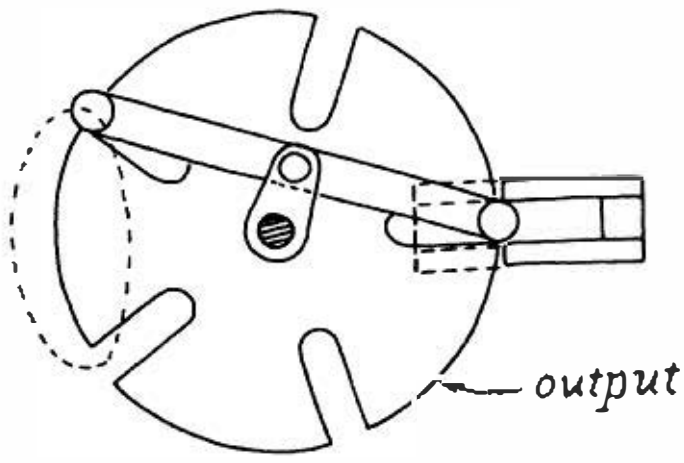
Drawing courtesy of PRODUCT ENGINEERING Magazine; Oct. 26, 1964: pp. 109, 110

Fig. 9–23. Groove cam Geneva.

modify the acceleration and deceleration characteristics of the Geneva quite significantly.

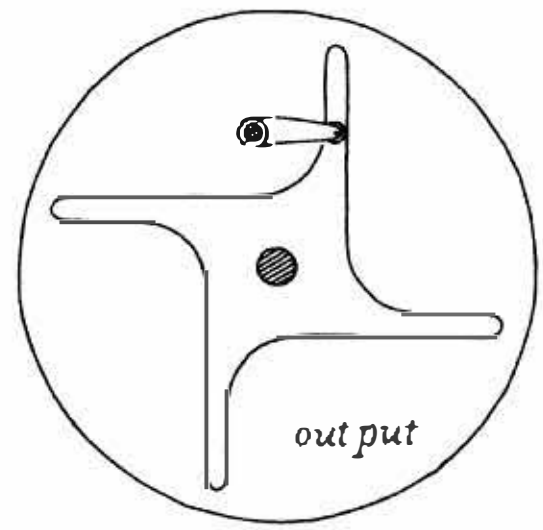
The locking slide Geneva, Fig. 9–24, is a variation of the four-bar drive system shown in Fig. 9–19. A pin on the slide crank engages the Geneva to lock the output member during dwell periods.

A modification of a four-slot internal Geneva is the internal groove Geneva shown in Fig. 9–25. With this design each roller exits and enters the driving slots tangentially, which is always desirable but sometimes difficult to accomplish with a conventional internal Geneva.



Drawing courtesy of PRODUCT ENGINEERING Magazine; Oct. 26, 1964: pp. 109, 110

Fig. 9–24. Locking slide Geneva.



Drawing courtesy of PRODUCT ENGINEERING Magazine; Dec. 20, 1965: pp. 86–88

Fig. 9–25. Internal groove Geneva.

Figure 9–26 is a drawing of an external Geneva driven by elliptical gears. The drive pins do not enter the Geneva wheel slots tangent to the center line, but pin velocity is at a minimum when they enter. And again, in the sketch to the right, the Geneva's motion curves can be seen to be modified drastically by this drive means.

In Fig. 9–27, a four-step sequence in the operation of a Geneva driven by an elliptical gear combination can be seen. It should be noted that the Geneva rotates 180 degrees in two steps for every revolution of the input.

Anatomy of the Geneva Mechanism

The designer should be completely familiar with the anatomy of a conventional Geneva and the

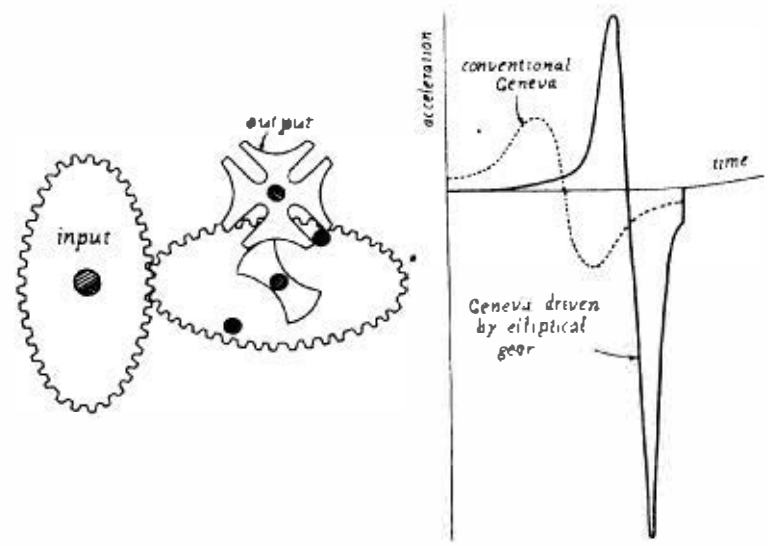


Fig. 9–26. (*Left*) External Geneva driven by elliptical gears. (*Right*) Comparative acceleration curves.

functions of its parts (Fig. 9–28)

A. Driver—rotates continuously; *B*. Geneva wheel (output)—rotates 90 degrees for every 360-degree motion of the input driver. It [illegible] a maximum acceleration of approximately 5 radians/sec^2 for each revolution per minute of input. *C*. Drive [illegible] hardened steel or equivalent and good fit in the drive slots of the Geneva wheel. Enters slot, moving parallel to the centerline of the slots for [illegible] "jerk." Cantilevered in most designs, yoke-mounted in some (supported at both ends) for maximum speed and load; *D*. Slot entrance—square, not chamfered, in best designs, where misalignment between drive pin and slot is eliminated by proper matching of *E* and *F*. Centerline of pin should be aligned with ends of slot when "contact" is first made; *E*. Locking ring on driver—hard, polished, smooth; used to hold driver during dwell periods. Must be good fit with *F* to guarantee proper location of drive slot-versus-drive pin at start of motion. *F*. One of four locking surfaces on Geneva wheel—the comments given for *E* apply here also; *G*. Bearings—should be sufficiently

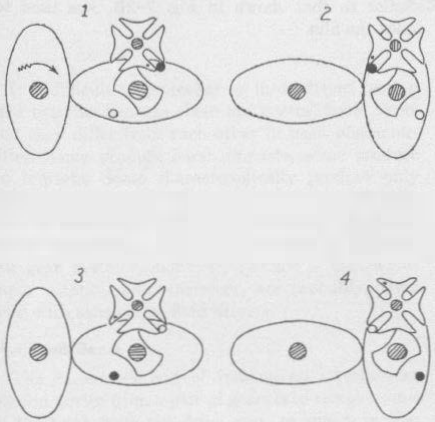
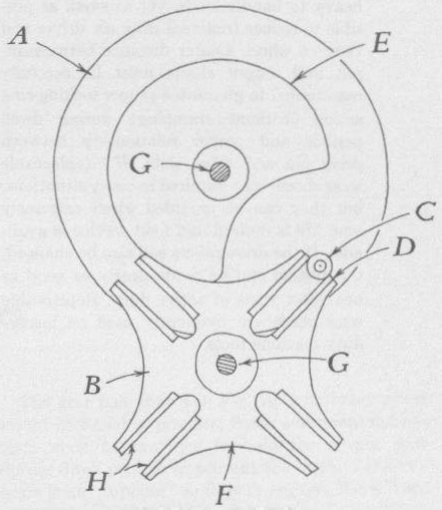
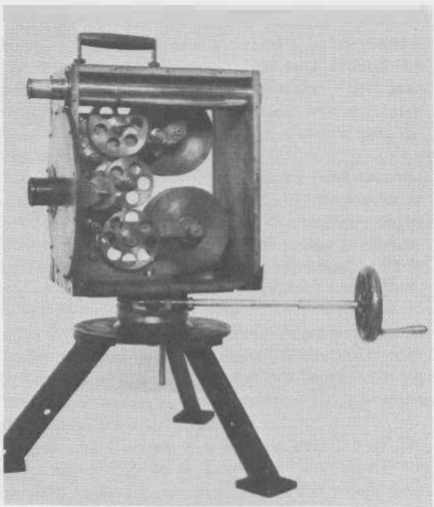


Fig. 9–27. Sequence of Geneva operation with elliptical gear combination drive.





British Crown copyright, Science Museum, London

Fig. 9–29. Antique cine-camera on a revolving stand, built in 1896 and used at Queen Victoria's Diamond Jubilee celebration. An early form of Geneva is used to index the film.

heavy to handle loads, yet as small as possible to reduce frictional drag on driver and Geneva wheel. Center distance between input and output shafts must be carefully maintained to guarantee proper locking-ring action (without cramping) during dwell periods and proper relationship between drive pin and drive slots. *H*. Replaceable wear shoes—not required in many situations, but they can be provided where extremely long life is desired and field service is available. If the drive rollers can also be changed, the system can be made nearly as good as new after years of active duty. Replaceable wear shoes are frequently used on heavy-duty machine tools.

Historical Note

Intermittent motion mechanisms are used to index film in cameras, projectors, film-processing equipment, etc. A wide variety of mechanisms has been used in photographic equipment right from the beginning. Such things as epicyclic gears, mutilated gears, and Genevas can be found in cameras and projectors built in the late 1800's. The photograph in Fig. 9-29 (British Crown copyright, Science Museum, London), is of Paul's Cine-camera on a revolving stand, which was built in 1896. A note on this print points out that this camera was used for the Diamond Jubilee; the celebration of the 60th year of Queen Victoria's reign. A Geneva mechanism, similar to that shown in Fig. 7–2B, was used to index the film.

CHAPTER 10

Gears

It is difficult to generalize on intermittent motion gear systems because there are several basic types and they differ from each other in most characteristics. Some produce harsh impacts, some produce no impacts. Some characteristically produce only instantaneous dwell, others are used because they produce dwells of very long duration. Some are simple and cheap, others are complex and costly. All gear systems, however, require a continuous input rotation and, therefore, are probably never used with solenoid or fluid drivers.

Mutilated Gears

The easiest method of making an intermittent motion device from a pair of gears is to remove some of the teeth from the drive gear, as shown in Fig. 10-1. A gear with some of its teeth removed is sometimes called a mutilated gear, and if it is operated at high speeds or under high-load conditions, its name is well deserved! But if operated under a light load or at a reasonable speed, this is a very simple and versatile intermittent motion mechanism. Dwell and motion periods can be varied quite a bit depending upon the sizes of the two gears involved. The velocity ratio between input and output can easily be controlled by selection of the proper gear ratio. Nevertheless, this mechanism is an impact producer, as shown in Fig. 10–2. This is a simplified picture from that which would be obtained in most practical design situations where elasticity in the various parts of the system would result in multiple rather than single impacts.

The gear pair shown in Fig. 10–1 probably never would be found in practice. Some additional mechanism must be provided to hold the output gear during dwell periods to prevent the teeth of the two gears from "topping" as they re-engage. We will see some ways of accomplishing this in later illustrations.

Cycloidal Gearing

Another type of intermittent motion gearing is shown in Fig. 10–3. This is called a hypoeycloidal gear train. A small "planet" gear runs around the inside of a fixed ring gear. A pin in the planet gear engages a slot in an output crank which rotates about the center of the ring gear, dwelling periodically as it rotates.

If the planet gear revolves around a fixed external or spur gear, as shown in Fig. 10–16, the system is called an epicycloidal gear train. The performance of these two types of cycloidal gear trains is similar. Anything we say about the hypocycloidal train is true or nearly true for the epicycloidal train, thus, we will discuss only the former.

The hypocycloidal gear train produces no impacts. It also produces only very short (theoretically only instantaneous) dwells. In these two characteristics, therefore, it differs significantly from the mutilated gear system considered earlier.

Figure 10–4 shows a schematic drawing of the gear train of Fig. 10-3. The dotted line represents the path taken by the drive pin as the planet gear moves around inside the ring gear. Since the diameter of the ring gear is an even multiple of the pitch

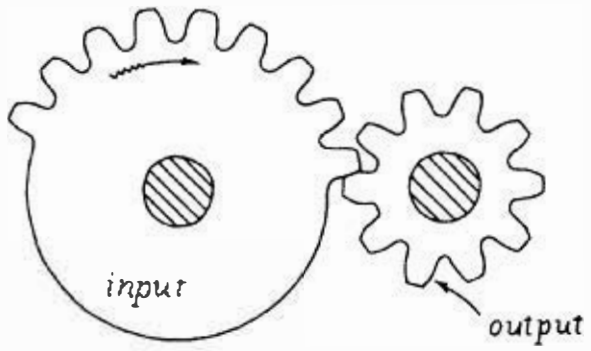


Fig. 10–1. Simple mutilated gear pair.

diameter of the planet gear (in the illustration, four to one) the drive pin will always follow the same path.

The instantaneous tangential velocity of the drive pin with respect to the center of the ring gear will vary as the rotational motion of the planet gear about its center is added to, and subtracted from, the rotational motion of the input arm about the center of the ring gear. (See Chapter 1 for the step-by-step procedure for determining the pin's tangential velocity.) The tangential velocity of the drive pin about the center of the system will be at a minimum in the position shown in Fig. 10–4 (and at the other three "corners" of the "square" being described by the pin). With the pin mounted about halfway between the center and pitch diameter of the planet gear, as shown in Fig. 10–4, the tangential velocity of the pin will never reach zero. If the pin were mounted on the pitch line of the planet gear, however, as shown in Fig. 10–5, it will describe a slightly different path, as shown, and there will be four times during each revolution of the input when

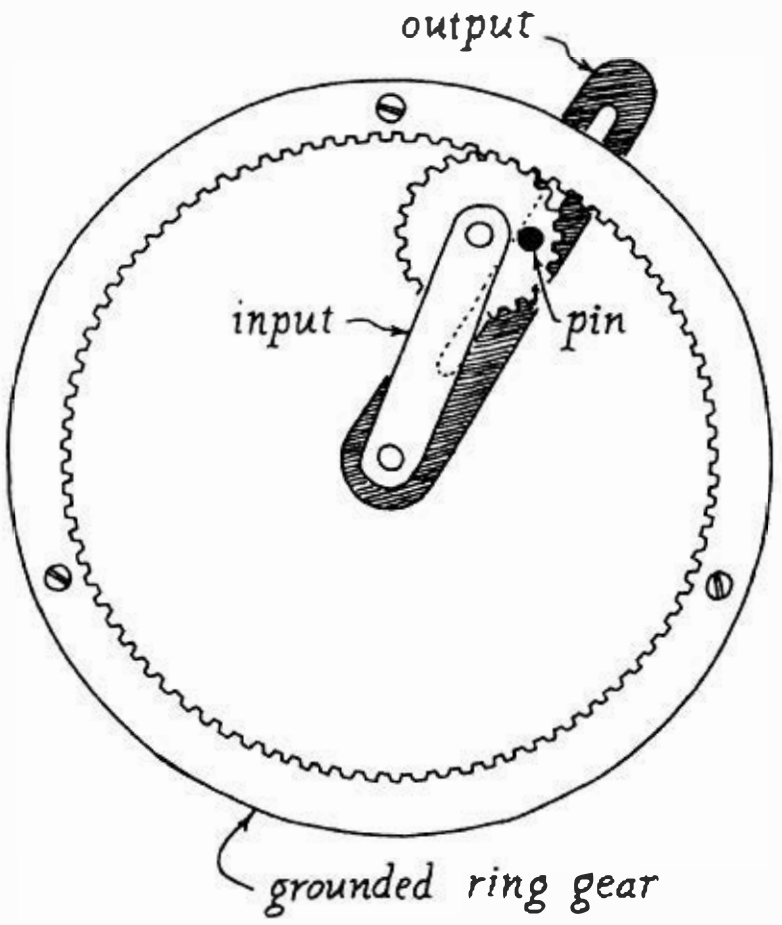


Fig. 10–3. Hypocycloidal gear train with a four-to-one ratio. The drive pin is located about halfway between the center and the pitch line of the planet gear.

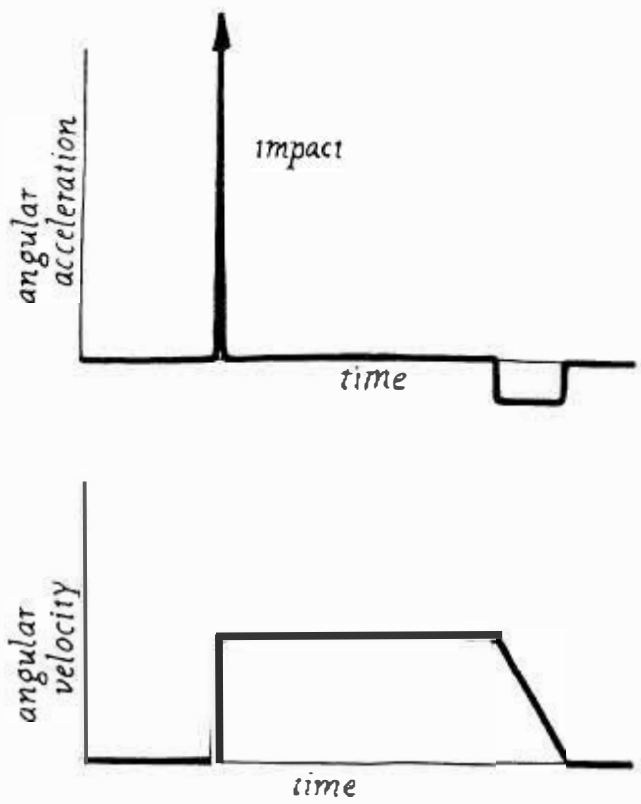


Fig. 10–2. Motion curves for the mutilated gears of Fig. 10–1.

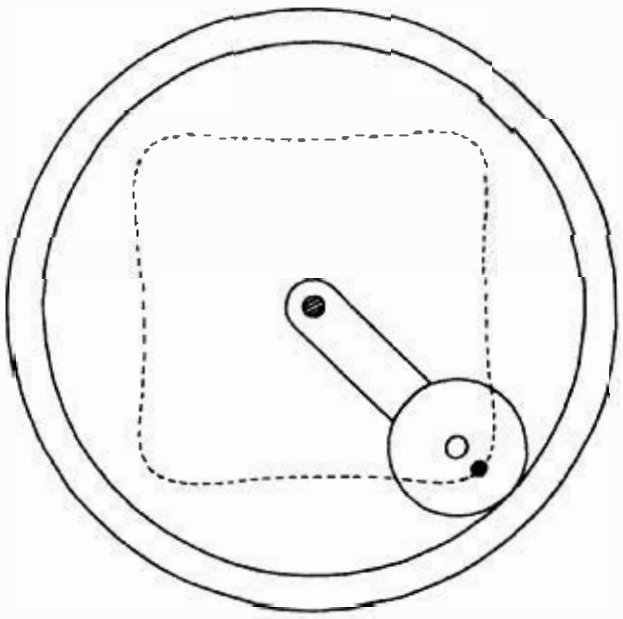


Fig. 10–4. Schematic illustration of the hypocycloidal gear train of Fig. 10–3.

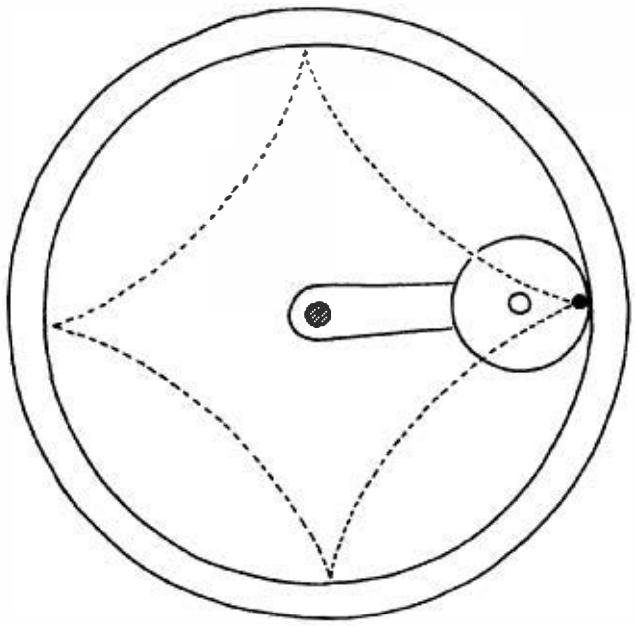


Fig. 10-5. Schematic representation of a four-to-one hypocycloidal gear train with the drive pin located on the pitch line of the planet and ring gears.

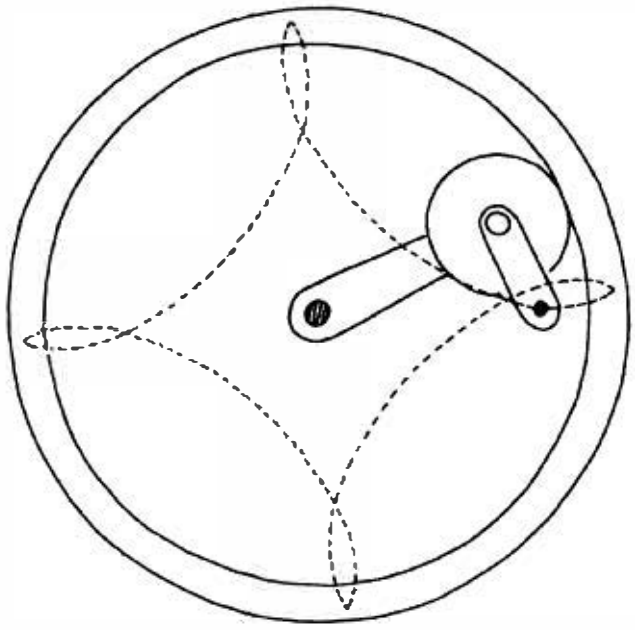


Fig. 10-6. Schematic representation of a four-to-one hypocycloidal gear train with the drive pin mounted beyond the pitch line of the planet gear.

the tangential velocity of the pin will be exactly zero. Theoretically, these are only instantaneous points but with backlash between planet gear and ring gear and/or backlash between the drive pin and the slot in the output crank, this instantaneous dwell period can be extended slightly. Of course, if it is extended too much by the introduction of backlash, impact can result when motion recommences.

If the drive pin is mounted on a small arm which is fixed to the planet gear so that the pin is operating on a radius that exceeds the pitch radius of the planet gear, then the pin will describe the motion path shown in Fig. 10-6. Under these conditions the tangential velocity of the pin around the center of the system actually reverses momentarily, four times during each drive cycle, as shown by the four cusps on the dotted-line path of Fig. 10-6.

Again, if backlash is purposely introduced into the system, perhaps between the pin and the slot in the output crank, then the entire period described by the cusps in the pin path can be converted to dwell in the output. Figure 10-7 shows the motion curves for the output crank of a hypoeycloidal system (for 180 degrees of rotation of the input crank) when the drive pin is located on the pitch diameter of the planet gear. As can be seen from the velocity curve the dwells are only instantaneous, as explained above.

Cycloidal gear trains can be used alone to produce intermittent motion if short dwell periods are acceptable. They are frequently used, however, with other intermittent motion mechanisms such as Genevas or ratchets: in this case, the cycloidal gear is included to reduce impact velocities in the other mechanisms. With a Geneva, for example, the cycloidal gear train would be used to reduce the velocity of the Geneva drive pin at the moment the

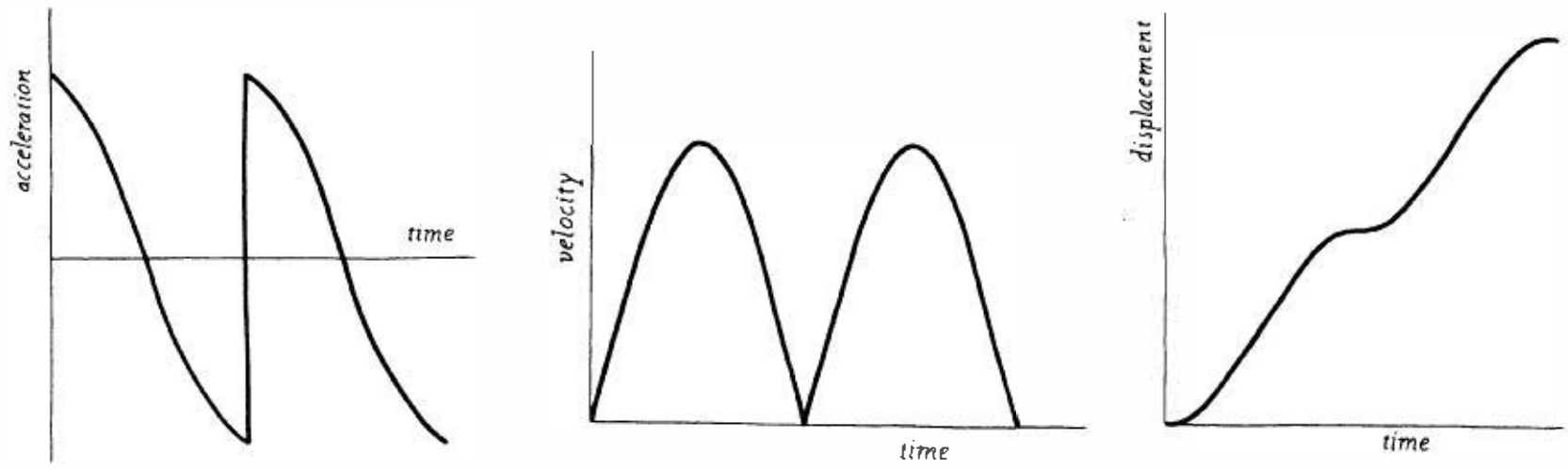
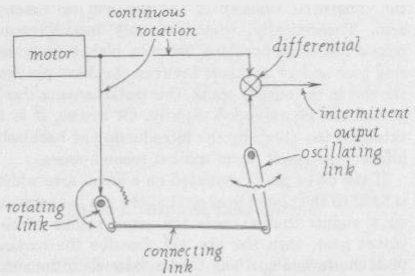
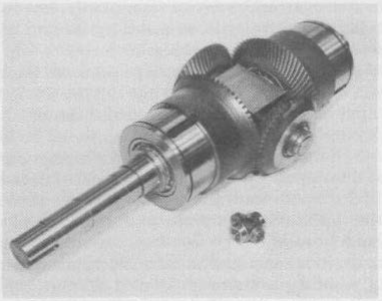


Fig. 10-7. Motion curves for a hypocycloidal gear train in which the pin is mounted on the pitch line, as in Fig. 10-5. These three curves represent the motion of the output crank of the gear train (shown in Fig. 10-3) for 180 degrees of rotation of the input crank.





Photograph courtesy of the Fairchild Hiller Corp.

Fig. 10–8. (*Top*) Schematic describing a system where a differential gear can be used with a motor and a four-bar linkage to produce intermittent motion. The differential is used to add a continuous motion to an oscillatory motion. (Also see Fig. 10–9). (*Bottom*) Two typical bevel gear differentials.

pin enters the slot of the Geneva wheel. We saw an example of this in Chapter 9

One advantage of cycloidal systems, of course, is the elimination of impact. A big disadvantage is the fact that only momentary dwells can be obtained These are not very popular intermittent motion devices, perhaps because of the short dwell or because they involve sliding parts, or simply due to the fact that many designers are not familiar with these devices

Differential Gear Systems

Differential gears are commonly used to add or subtract one shaft rotation to or from another Strictly speaking, they are also cycloidal gears, but we will discuss them separately because they (we can utilize) more than one input to obtain produce a variety of outputs. Taken alone differentials are incapable of producing intermittent motion. With the proper inputs, however, they can produce intermittent motion without impact. In Fig 10–8, for example, we see (top, a schematic of a system in which a motor continuously drives one side of a differential such as the bevel gear differential in the photograph, and also drives the input of a four-bar linkage. The oscillatory output of the four-bar linkage is fed to the other input of the differential, where it periodically adds to and subtracts from the steady input from the motor. The result of these continuous and momentary inputs is intermittent output motion with momentary dwells, as shown by the velocity curves of Fig. 10–9. Again, dwell is (theoretically) only instantaneous, but with backlash, etc., inthis system it will probably be longer than instantaneous in any practical situation. If number 2 in the differential system were provided with a reversible electric motor drive of its own, the dwell period could be anything the designer wanted to make it. This is a big advantage of the differential approach. The differential in this case is

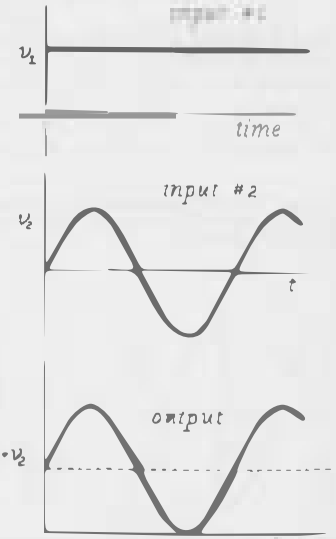
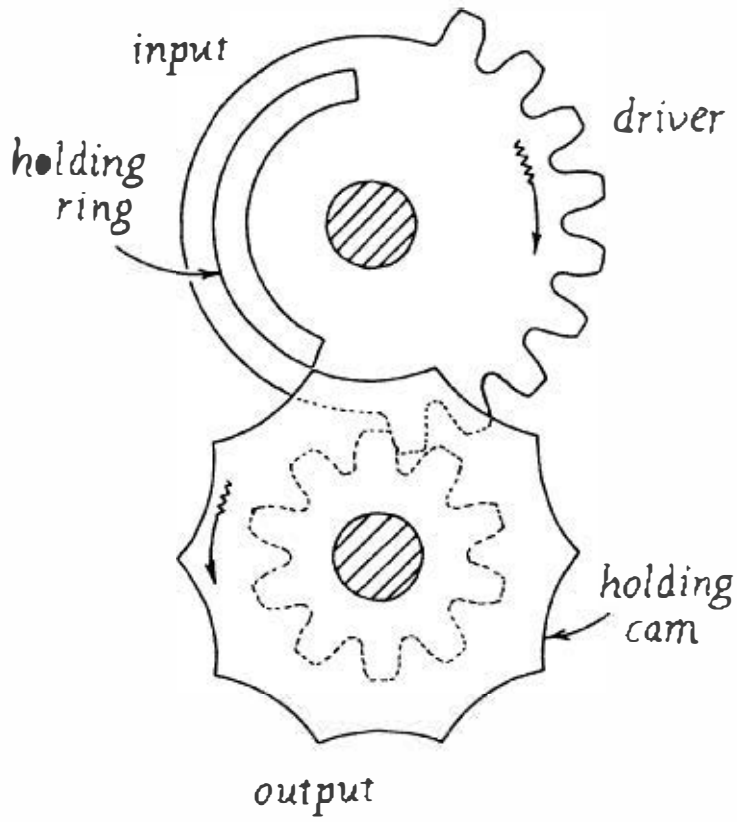


Fig. 10–9. Curves showing the input and output velocities of the differential system shown schematically in Fig. 10–8.

really replacing one or more clutch-brake combinations.

The designer should always be sure that his differential is designed to dissipate the amount of energy that it will have to absorb if it is subtracting the total input received from one electric motor, from the total received from another. When the output of the differential is zero, all input work must be dissipated within the differential itself. This "circulating horsepower" as it is called by gear designers, can produce heat and wear even when the system is apparently doing no work.



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 10–10. Mutilated gear system with locking or holding ring and holding cam.

Miscellaneous Gear Systems

Mutilated, cycloidal, and differential gears are the main types used to produce intermittent motion. Special situations demand special solutions, however, and clever designers have thought up other unusual ways to use gears to produce stepping outputs. We will take a look at some of these after we have considered several versions of the traditional methods.

A mutilated gear system with locking or holding ring and holding cam to lock the output gear during dwell periods is shown in Fig. 10–10. Some such holding arrangement is essential in the mutilated gear system to eliminate topping of teeth as the gears engage. Topping, of course, could cause impact, jamming, complete destruction of the teeth, etc., and must be prevented.

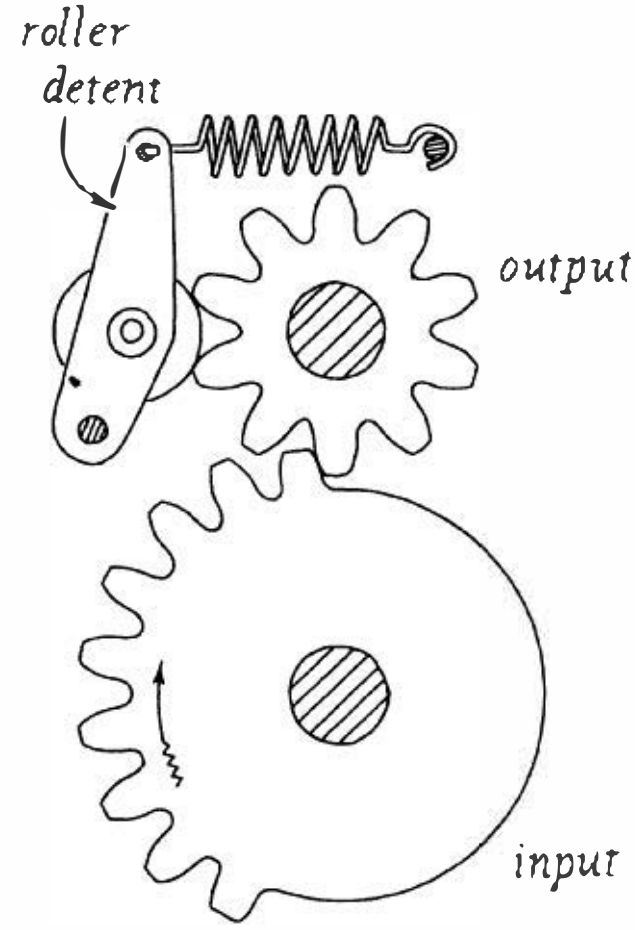
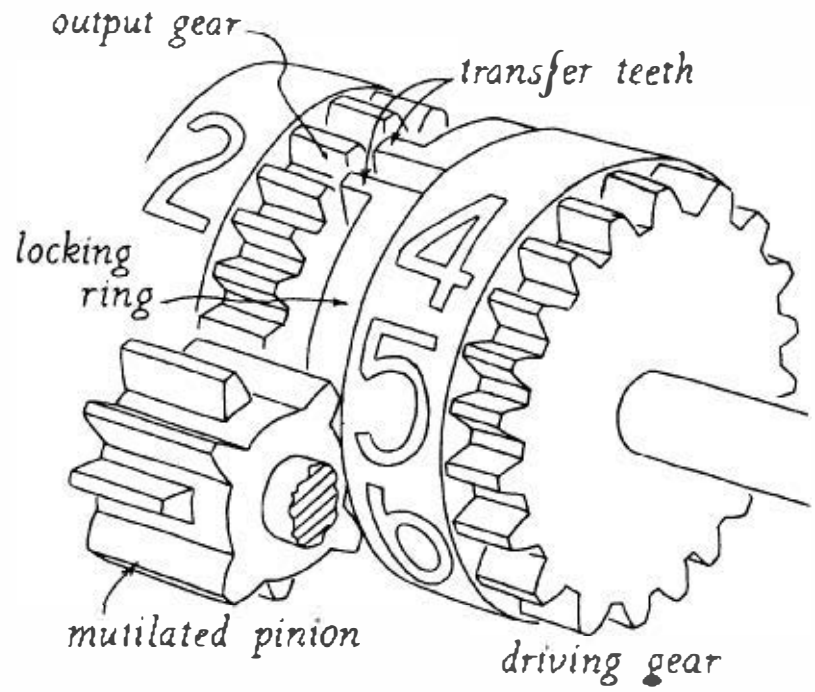


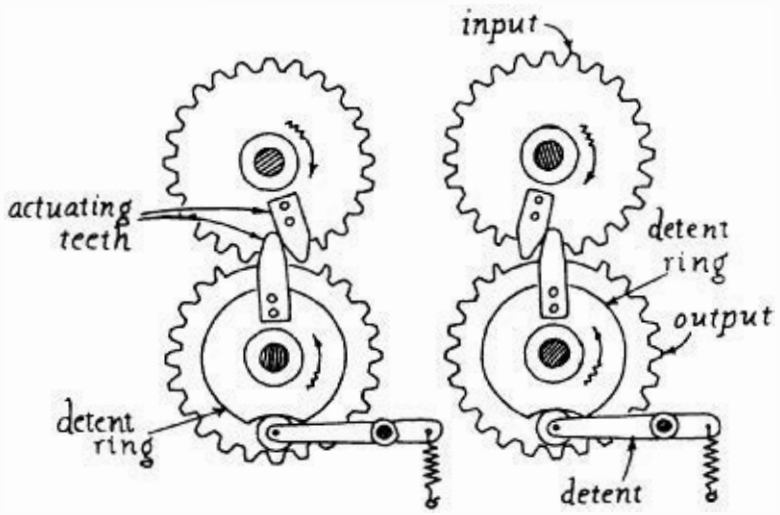
Fig. 10–11. Mutilated gear with roller detent.

Another method of holding output during dwell periods, perhaps not quite as common a method as that shown in Fig. 10–10, is the mutilated gear with roller detent (Fig. 10–11), used in place of a locking ring.



Drawing courtesy of the Veeder-Root Company

Fig. 10–12. Mutilated pinion transfer mechanism used in a mechanical counter.



Drawing courtesy of PRODUCT ENGINEERING Magazine; Dec. 20, 1965; pp. 86-93

Fig. 10-13. Mutilated gear arrangement with large actuating teeth.

Figure 10-12 shows a mutilated pinion transfer mechanism used in a mechanical counter. The alternate teeth on the pinion are extended full length to engage a locking ring on the drive wheel (the right-hand number wheel). The effect is the same as that obtained by the holding ring and holding cam surfaces of Fig. 10–10, however, the embodiment is different.

Some teeth are also missing from the output gear of the mutilated gear arrangement shown in Figure 10–13, but here large actuating teeth are provided to initiate motion. Note that the output is also locked by a roller detent rather than by a locking ring. The ratio of motion to dwell differs from that obtained with a conventional mutilated gear.

There will be two dwell periods and two motion periods for each revolution of the input shaft in the mutilated gear system, as shown in Fig. 10-14. One of these motion periods will be twice the velocity of

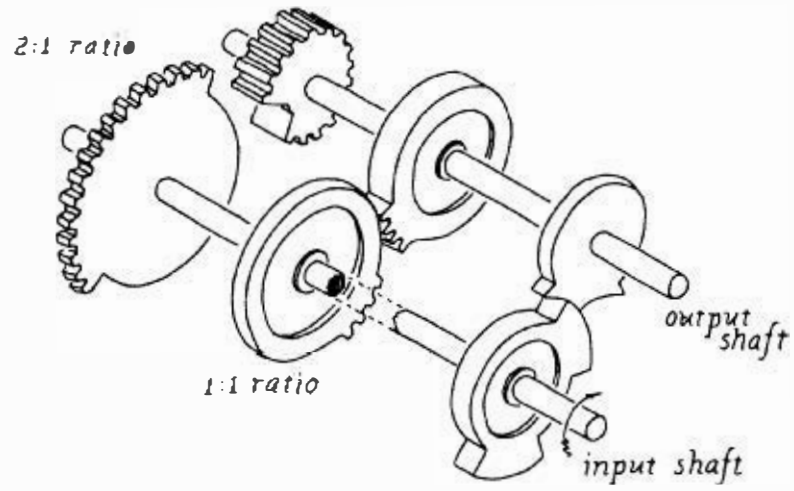


Fig. 10–14. Mutilated gear system with output motion at two different speeds.

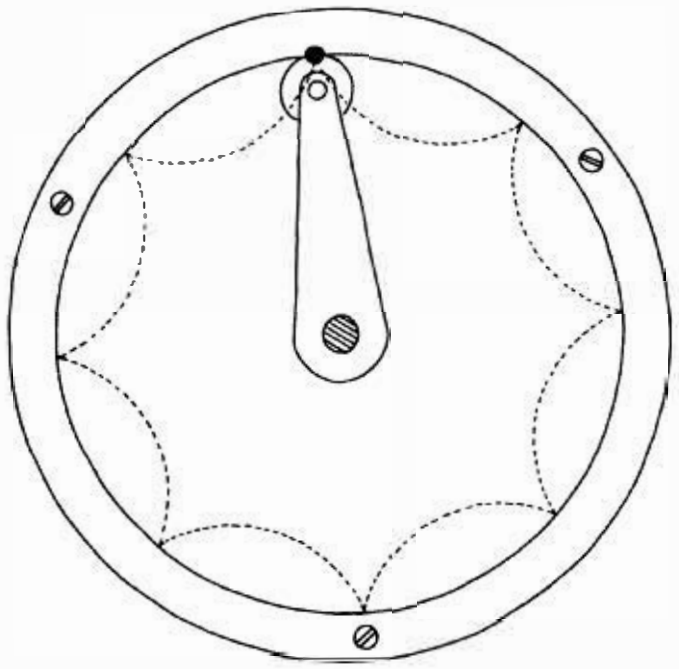
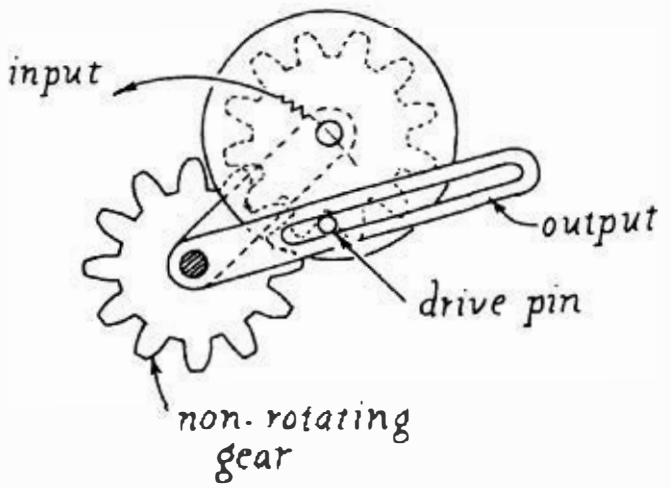


Fig. 10–15. Hypocycloidal gear train with an eight-to-one ratio.

the other; as first the right-hand and then the left-hand mutilated gear pairs will drive the output.

This hypocycloidal gear train system (schematic, Fig. 10–15) with an eight-to-one ratio and the drive pin located on the pitch line of the two gears, will produce eight instantaneous dwells per revolution of the input arm. Compare this system with that in Fig. 10–5, which has a four-to-one ratio.

Epicycloidal trains produce momentary dwells similar to those produced by hypocycloidal gear trains. The system shown in Fig. 10–16, an epicycloidal gear train with a one-to-one ratio, would produce one dwell per revolution of the input crank. As with hypocycloidal gears, the sun gear diameter must be an even multiple of the planet gear diameter



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 10–16. Epicycloidal gear train with a one-to-one ratio.

if the dwell is to occur at the same point in each rotation.

The schematic shown in Fig. 10–17 is of a two-to-one ratio epicycloidal gear train with the drive pin located on the pitch radius of the planet gear. Such a system would produce two dwells per input revolution as shown.

Figure 10–18 shows another design in which an oscillatory motion is added to, and subtracted from, a continuous motion. This is a planetary gear train in which the motion of a segmented planet gear is modified by a cam.

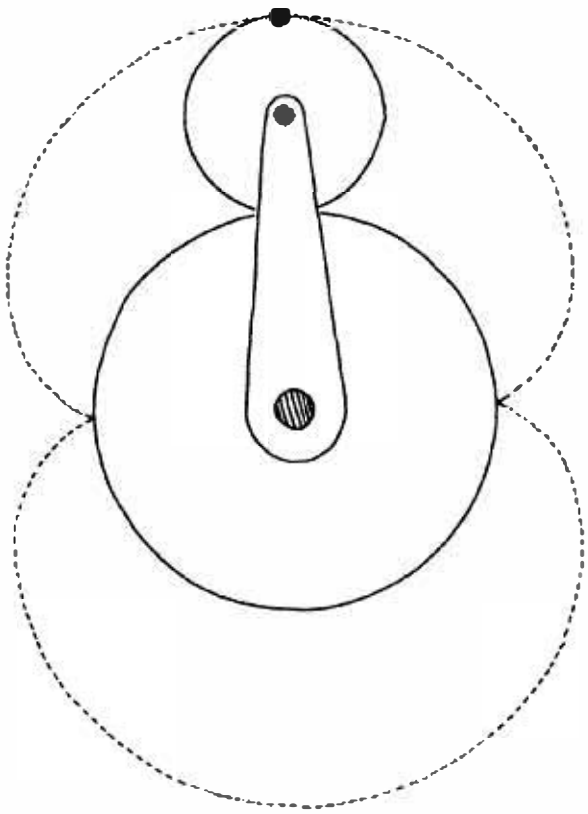
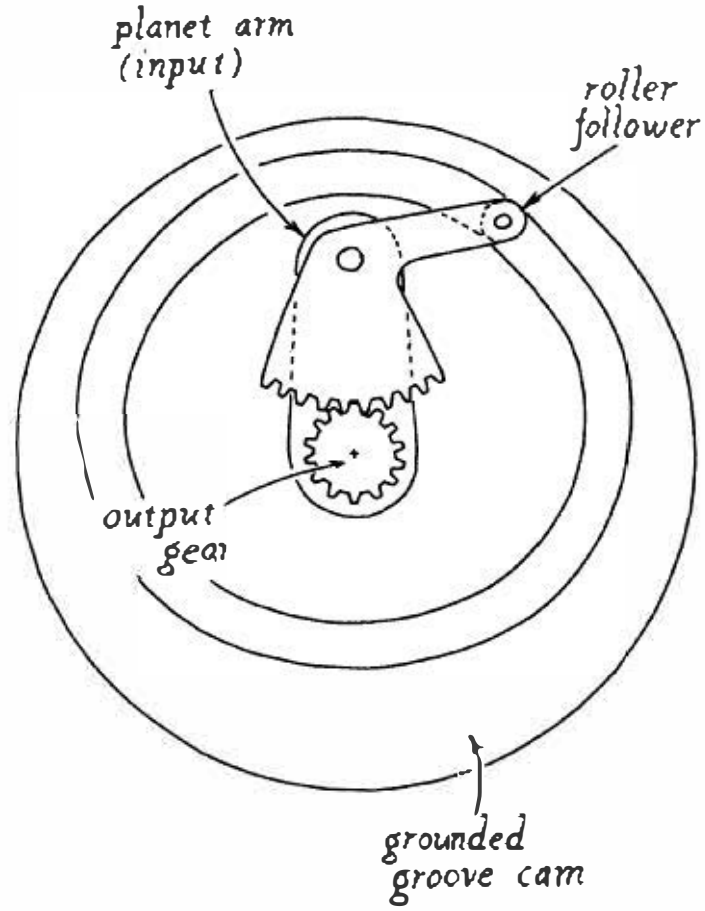


Fig. 10–17. Two-to-one ratio epicycloidal gear train with drive pin located on the pitch radius of the planet gear.

Noncircular gears modify the motion of the planet gear in the planetary gear mechanism shown in Fig. 10–19, to produce intermittent output from a single continuous input. The ratio of motion to dwell time can be adjusted by varying the shapes and diameters of the gears. Output motion is zero in a planetary gear assembly when the inputs cancel each other. With these noncircular gears, one input cancels the other periodically during each cycle. This occurs when the radius of gear (1) equals that of gear (2); and, at the same time, the radius of gear (3) equals that of gear (4). (See also detailed drawing in Fig. 10–19.)

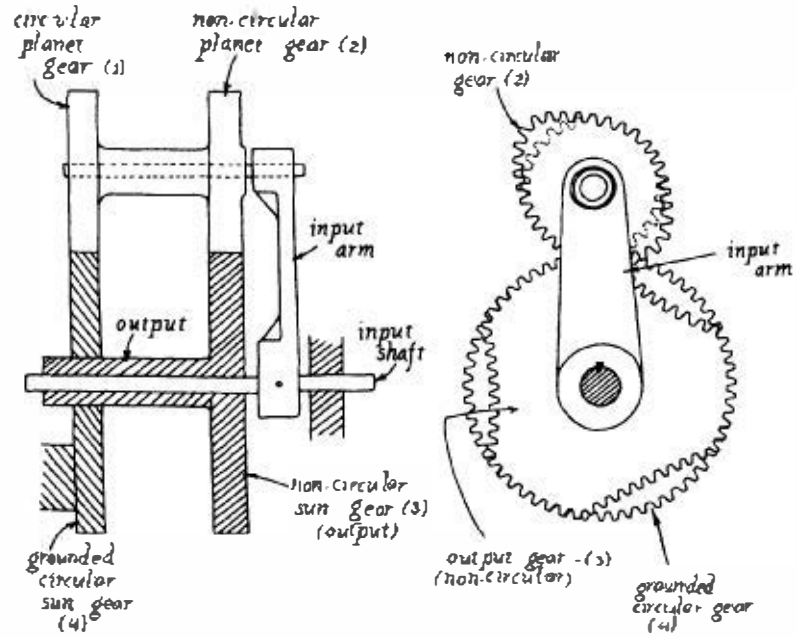
As with other cycloidal gear systems, the length



Drawing courtesy of PRODUCT ENGINEERING Magazine; Jan. 4, 1960; pp. 58, 59

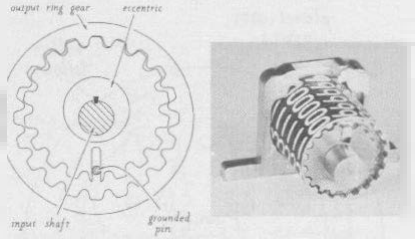
Fig. 10–18. Planetary gear train. Segmented planet gear motion is modified by a cam.

of dwell of this simple hypocycloidal gear mechanism (*left*, Fig. 10–20) used in a small counter (*right*), will depend upon the backlash between the grounded pin and the slot in the input gear.



Drawing courtesy of PRODUCT ENGINEERING Magazine; Nov. 27, 1971

Fig. 10–19. (*Left*) Planetary gear mechanism with non-circular gears. (*Right*) Details.



Photograph courtesy of the Vander-Root Company
Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 161 ff

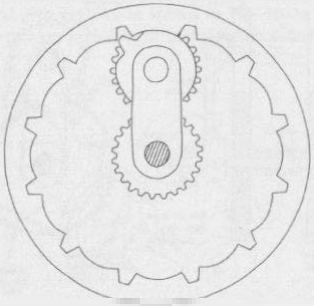
Fig. 10–20. (*Left*) Simple hypocycloidal gear system used in a small countermechanism. (*Right*) Small counter mechanism.

In this design, Fig 10-21, the cycloidal gearing is used to reduce the velocity of the drive pin as it enters the drive slots in an internal Geneva wheel. This converts the short dwell of the cycloidal train to the relatively long dwell of a Geneva mechanism.

In a modified motion Geneva system (Fig. 10–22) gears are used to modify the motion of a Geneva drive pin, the goal again being to reduce the impact velocity of the pin as it enters the slot.

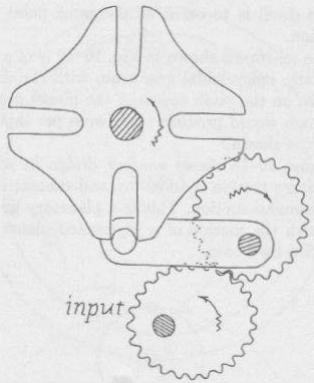
Systems such as the one shown in Fig 10-23 could be used with motors, clutches, brakes, linkages and other devices to produce a wide variety of intermittent motions for heavy-duty applications.

Figure 10–24 is a copy of the patent drawing for a single-input differential gear assembly. In this clever,



Drawing Courtesy of PRODUCT ENGINEERING [illegible] 1964 pp 109, 110

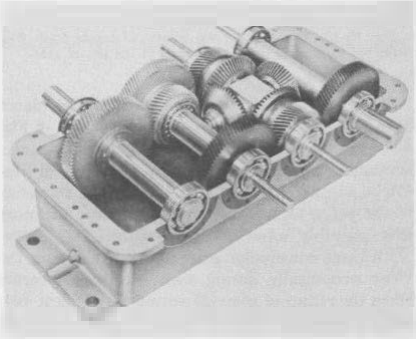
Fig 10–21. Cycloidal gear Geneva system.



Drawing courtesy of PRODUCT ENGINEERING M[illegible] 1964 pp 67 68

[illegible]

noninpacting mechanism developed at the National [illegible] Company, a single [illegible] to a differential system produces an intermittent output. The [illegible] is through shaft *12* and the ring gear *38*, is fixed (grounded). The output of the system is on shaft *18*. Perhaps the system can be best understood by reference to the explanatory details in Fig. 10-25. The [illegible] at the top of this [illegible] produce rotation in the output shaft whenever the input shaft is rotated. Motion would be on a



Photograph courtesy of the Fairchild Heller Corp.

23. Large differential gear assembly with the cover removed.

one-to-one basis. If the bevel gears in the bottom sketch are rotated, they will drive the output shaft through the crank and yoke mechanism, producing oscillatory output. Now referring back to Fig. 10–24; as the input shaft (12), is rotated, it in turn rotates yoke *16*. This would produce pure rotation in output shaft *18* if it were not for the fact that the bevel gears *40* and *44*, are forced to rotate by the action of the fixed ring gear *38*, as the yoke *16*, rotates. As a result, oscillatory motion is added to and subtracted from continuous motion (as was seen in Fig. 10–9) to produce an intermittent output with (theoretically) instantaneous dwells.

Here, in Fig. 10–26, is another version of the differential mechanism patent shown in Fig. 10–24. Shaft *48* and shaft *30* are continuously driven by separate

Nov. 4, 1969 J. M. STEINKE 3,475,977

ROTARY STEP MECHANISM

Filed Jan. 2, 1968 3 Sheets-Sheet 1

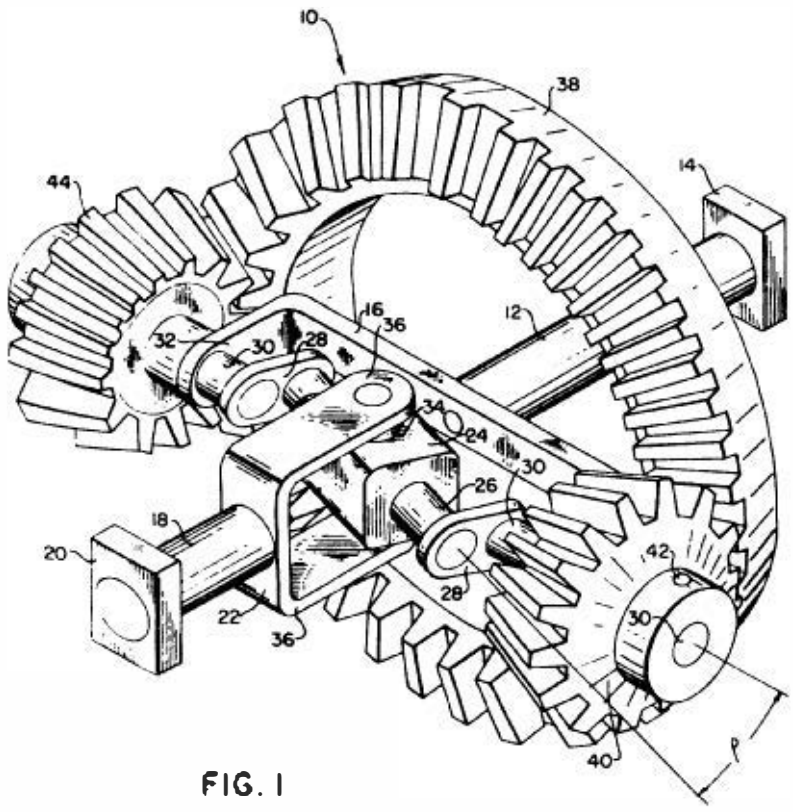


FIG. 1

INVENTOR
JAMES M. STEINKE
BY Louis A. Klein
Albert L. Sessler, Jr.
Emerson, Fargo
HIS ATTORNEYS

Fig. 10–24. Rotary step mechanism, patent sheet 1. (U.S. Patent 3,475,977; J. M. Steinke.)

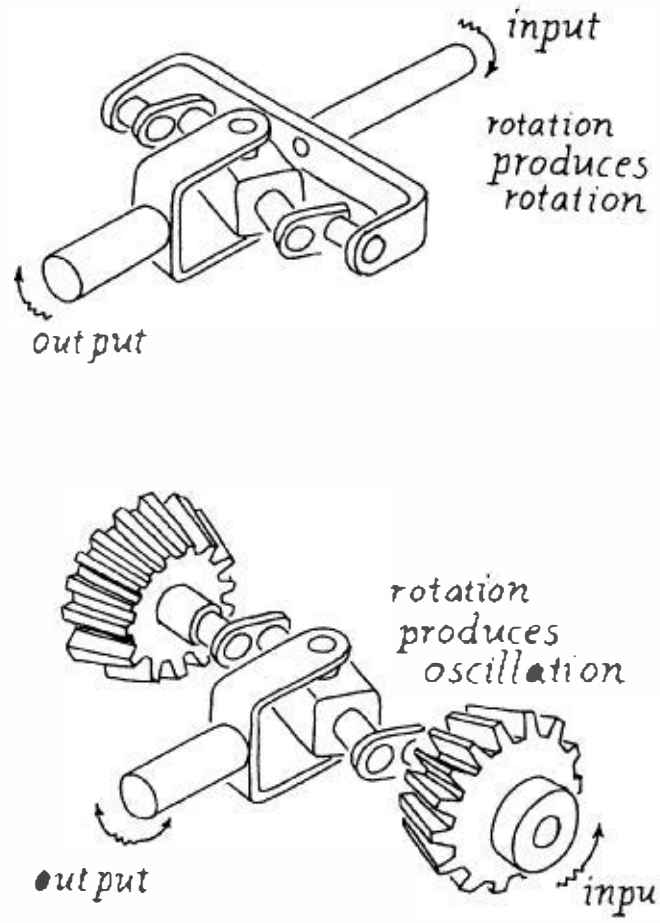


Fig. 10–25. Some analysis of the rotary step mechanism in Fig. 10–24.

motors. Shaft *48*, driven alone, would produce continuous rotation in output block *56*; shaft *30* alone would produce oscillatory motion in the output. If both inputs are introduced at the same time, therefore, the output motion will be intermittent. The inventor claims that the system can be used to produce hundreds of dwells in the output for each revolution of output shaft *54*. Although the dwells are theoretically only instantaneous in this mechanism, the inventor also states that velocity in the output is low enough, for a long enough period at each dwell point, to make the actual dwell finite and useful.

A rack-and-gear system used on a heavy-duty indexing table is shown in Fig. 10-27. The input rotates continuously. In sketch (A), at the top of the illustration, however, the output gear is held stationary by rack gear #1. The drive gear then merely rolls along the surface of the output gear without producing any output motion. When it is desired to index the table, rack #1 is extracted from the output gear and, as shown in (B) lower sketch, rack #2 is moved to the right to engage the left-hand side of the drive gear. The drive gear now rolls along rack #2 and drives the output gear. The two gear racks could be controlled by a cam on the same input

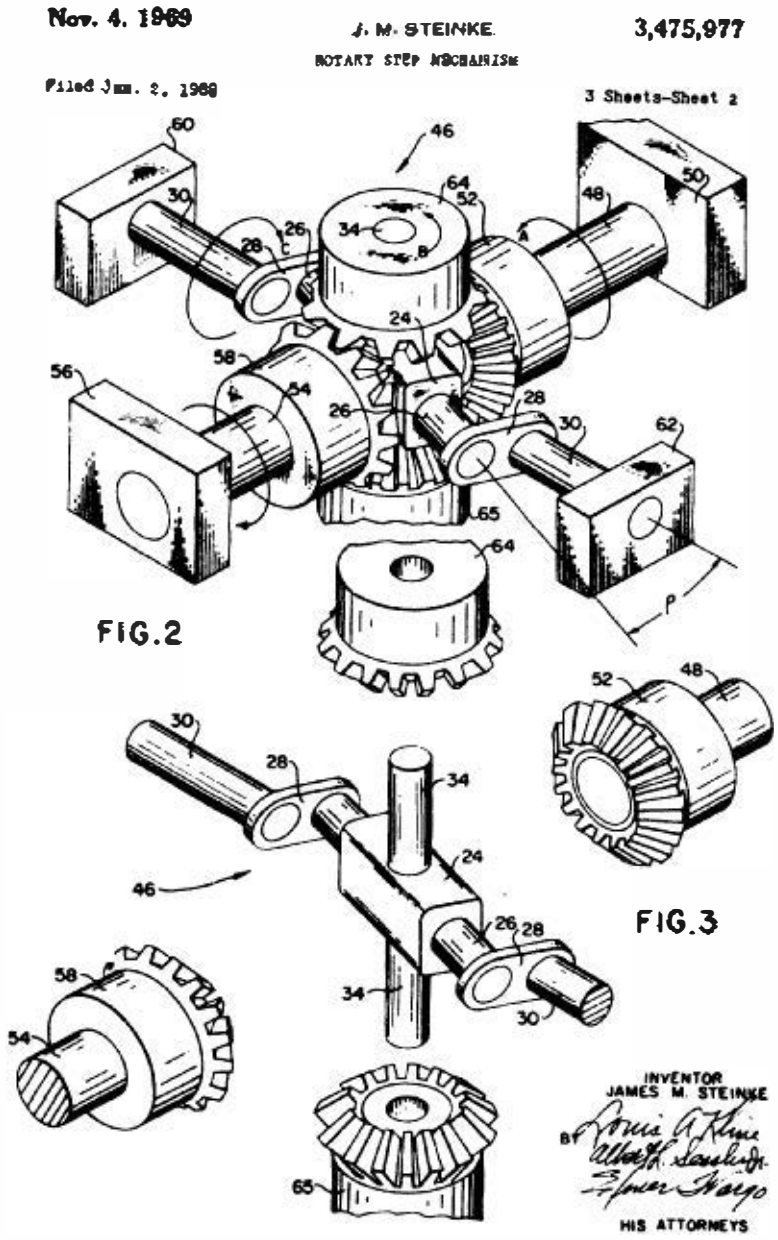


Fig. 10-26. Rotary step mechanism, patent sheet 2. (U.S. Patent 3,475,977; J. M. Steinke.)

shaft that moves the drive gear, to produce automatic indexing; or they could be driven separately to put the dwell period under the manual control of an operator.

In another interesting gear mechanism from National Cash Register (Fig. 10-28), an oscillatory motion is added to and subtracted from a continuous motion to produce intermittent motion output. Figure 10-29 illustrates this system with some detailed drawings. If the segmented gear shown in the mechanism at the top of the illustration would oscillate, it would produce oscillation in the output gear. If the ring in the center sketch of Fig. 10-29 is rotated, it will rotate the output gear on a one-to-one basis, thanks to the gear segment that is part of the ring. Thus, if a segmented gear is fastened to a ring, but is forced to oscillate as the ring rotates, it will produce intermittent motion in the output gear, as shown in the sketch at the bottom. In the mechanism of Fig. 10-28, the segmented gear is fastened to a pair of cam followers that force it to oscillate as the ring in which it is mounted is rotated around the output gear.

Earlier it was said that gears are always used in systems which have a continuously rotating input. However, every rule has its exception and Fig. 10-30 shows one, for the driver in this case is operated by a solenoid. The cam surfaces on the driver index a mutilated pinion identical to that in Fig. 10-12. The mutilated pinion, in turn, drives the output gear.

The three-gear system in Fig. 10-31 has to be seen to be believed. The input crank rotates through 360 degrees, carrying with it the left-hand gear which is rigidly fastened to the input crank. The output gear will either hesitate periodically, dwell instan-

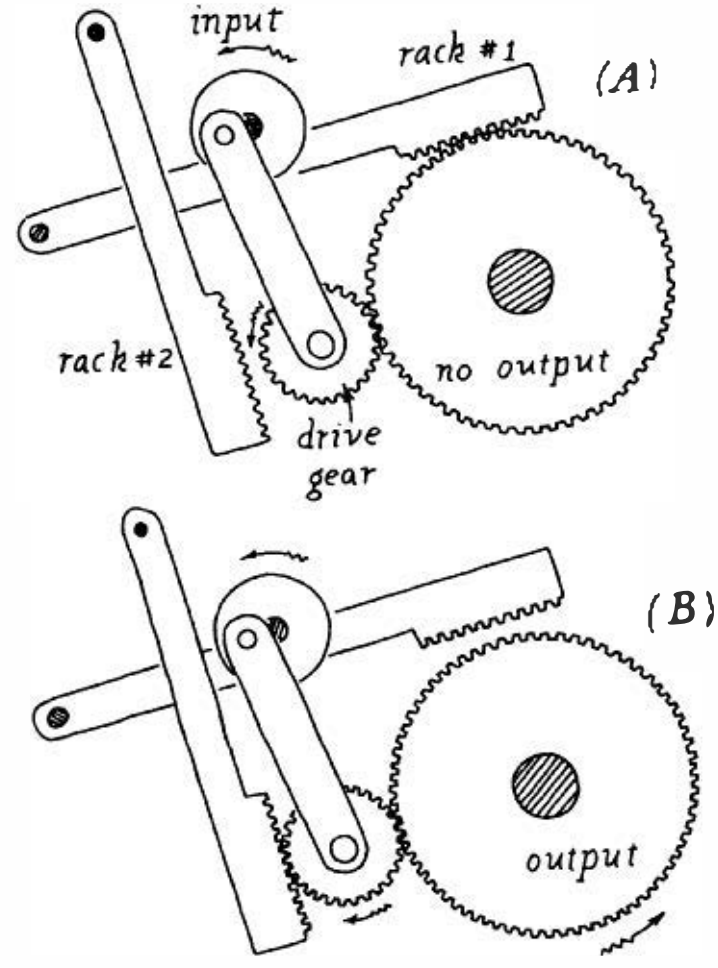


Fig. 10-27. Rack and gear system used on a heavy-duty indexing table. (Drawing made from information supplied by Gilman Engineering and Manufacturing Co.)

taneously periodically, or reverse motion during each revolution of the input crank depending upon the gear ratios involved.

The large input "gear" in the system shown in Fig. 10-32 has teeth (pins) which can be set in one of two positions. In the outer position they will

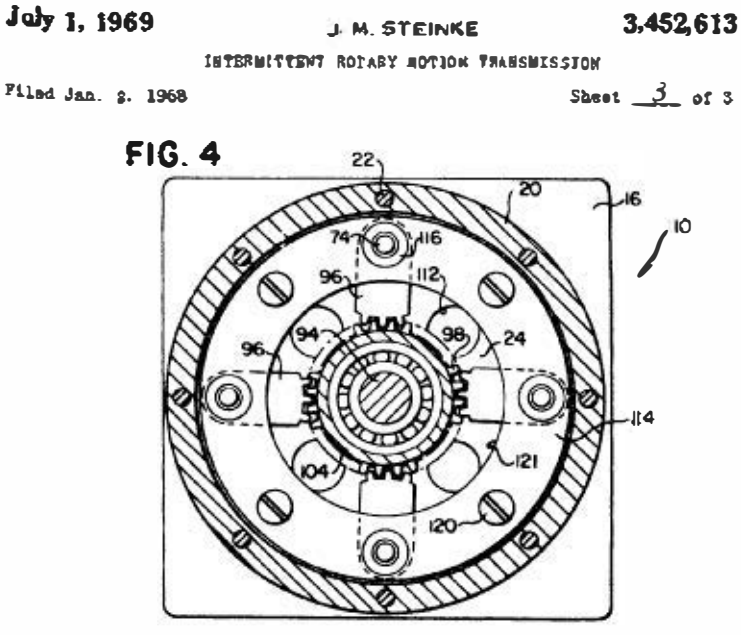


Fig. 10-28. Intermittent rotary motion transmission. (U.S. Patent 3,452,613; J. M. Steinke.)

engage the teeth of a small star wheel as the input disc rotates. If the pins are moved toward the center of the input disc they will not engage the star wheel. As a result, the output of the star wheel can be programmed to some extent. The model I saw had a 6-inch diameter input disc with 40 pins and a 1-inch diameter output wheel. Note that a detent roller is used to hold the output during dwell periods.

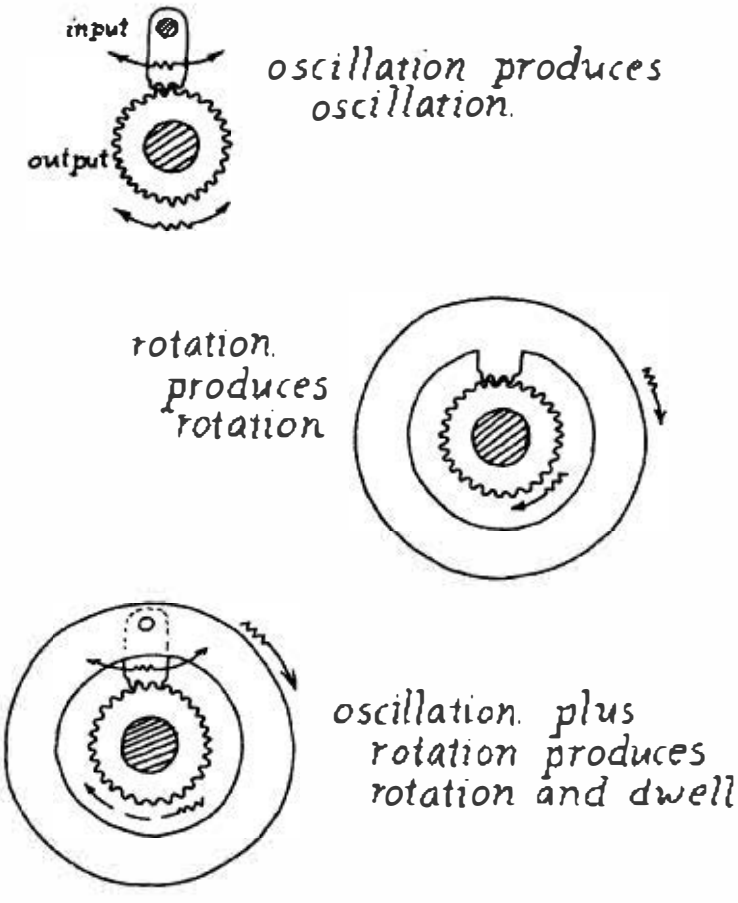


Fig. 10-29. Analysis and details of the gear mechanism of Fig. 10-28.

Anatomy of a Gear

There is a very great deal more to a gear than meets the unpracticed eye. The information below, on various parts of a gear, and shown in Fig. 10-33,

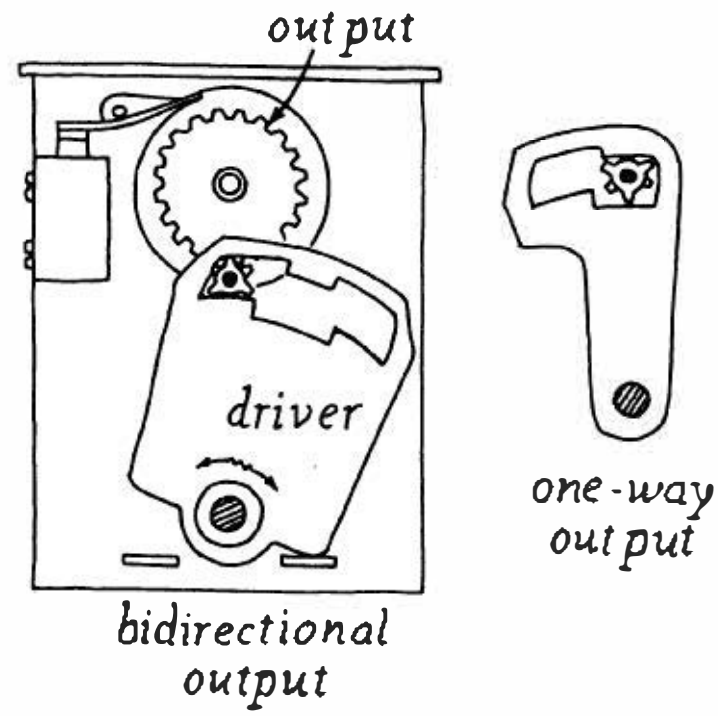
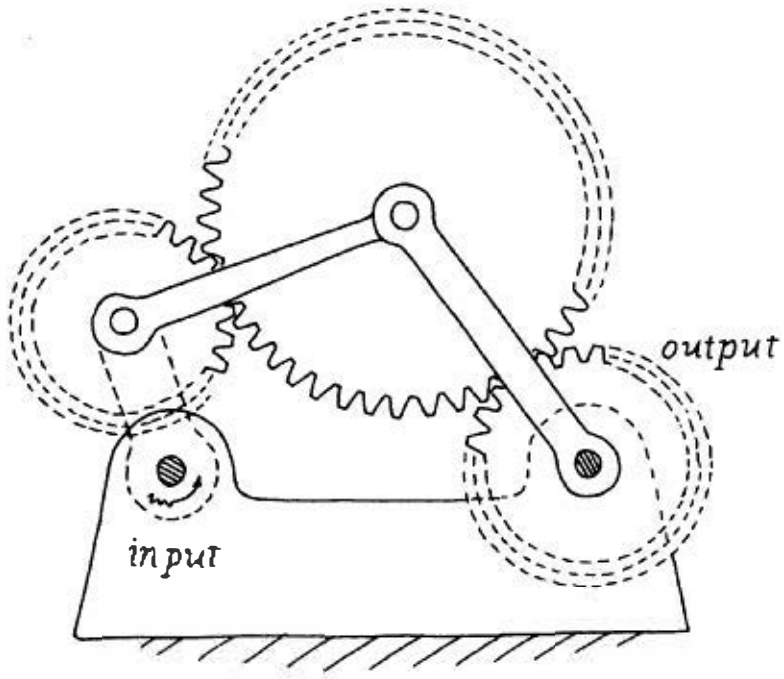


Fig. 10-30. Continuously rotating gear system with solenoid operated driver. (U.S. Patent 3,390,834; H. L. Seiden.)



Drawing courtesy of PRODUCT ENGINEERING Magazine; June 8, 1964; pp. 67, 68

Fig. 10–31. Three-gear system.

does not even come close to scratching the surface, but may, at least, show you that gears are not as obvious as they look. And these are "plain" gears; not bevel, worm, helical, herringbone, spiral, skew, hypoid, miter, hypocycloidal, or intermittent gears!

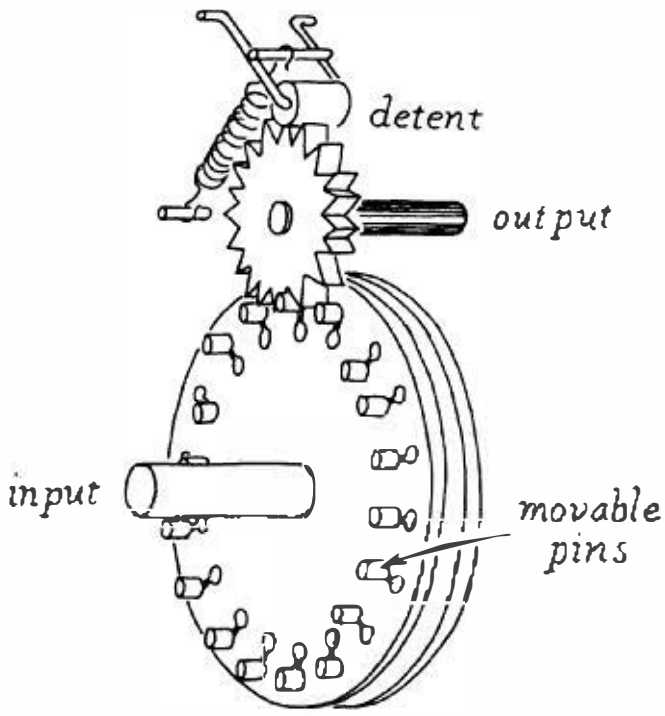


Fig. 10–32. Programmable "gear."

A. Drive gear, input—running at constant velocity; *B*. Driven gear, output—should also run at constant velocity; *C*. Single gear tooth—could be (and is) a variety of shapes. But once its shape is chosen, the shape of the mating gear tooth is determined by the requirement that the output gear "should also run at constant velocity." Of all the shapes chosen for gear teeth the most popular is the involute curve because it is relatively easy to generate; it will produce constant velocity in the output gear (if both input

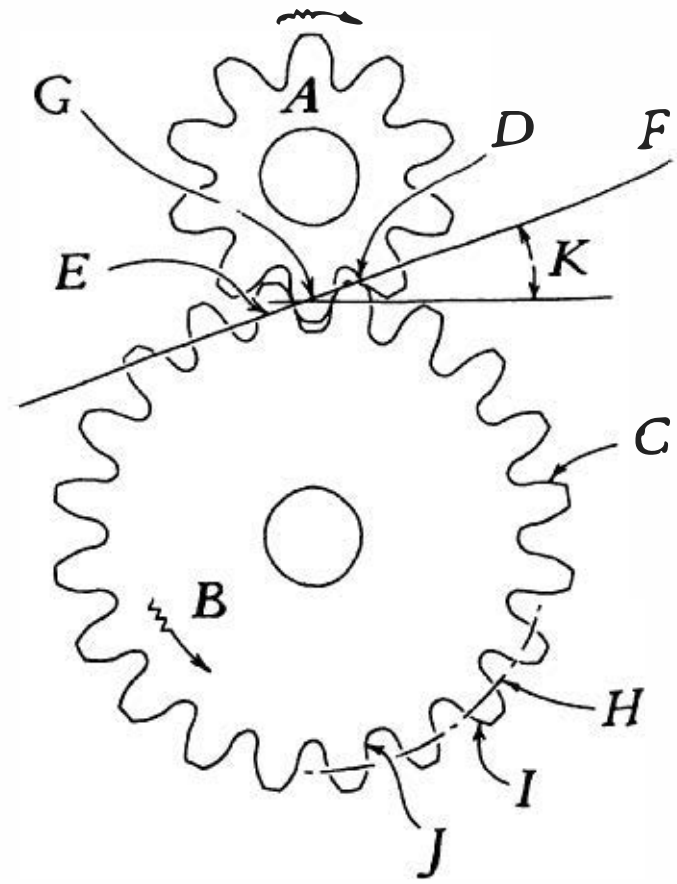


Fig. 10–33. Anatomy of a gear.

and output teeth are involutes); and it provides for interchangeability of gears, since input and output teeth are identical. Sometimes, the involute is modified slightly, however, to avoid undercutting of the teeth (for example on a small pinion); *D*. The point at which the teeth of the two gears first make contact as they turn; *E*. The point at which the teeth last make contact; *F*. Line of action of the gear teeth—joins *D* and *E*; passes through *G*. All contact between the gears must be (and nearly is) along this line if a

constant "lever ratio" is to be maintained between driving and driven gears at all times. The line is also tangent to the base circles of the involute curves of the teeth on both gears, *G* Pitch point—intersection of the line of action and of the line joining the centers of the two gears, *H* Pitch circle—might be considered the "centerline" of the teeth on both gears. Passes through the pitch point and is concentric by it) *I* Addendum—that portion of the gear tooth that is outside the pitch circle; *J* Dedendum—that portion of the tooth that is below the pitch circle. Usually the dedendum is made larger than the addendum to provide clearance for the mating teeth *K*. Pressure angle—angle between line of action and the line tangent to the pitch circles of the two gears and passing through the pitch point. A small pressure angle would be more efficient than a large, because conventional gears cramp each other for that portion of tooth contact that lies in front of the line joining their centers, and a smaller pressure angle results in less cramping. But a small angle also results in undercutting of the teeth and a subsequent loss of strength. The most common pressure angles are 14½ degrees and 20 degrees.

Mechanical Fuses

Several of the gear systems we have seen in this chapter were designed for high-speed business ma-

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Fig. 10-34. [illegible] ratchets to move the platen and machine escapements to index the carriage.

chines, intermittent motion and systems of all types have been, and are being used, in such machines, of course. In fact, this is probably the area in which the most intermittent motion development work is being done at the present time. How to move paper, cards, printing heads, etc., fast enough to keep up with [illegible] electronics. It is quite a challenge. The photograph in Fig. 10-34 shows a 1907 Remington typewriter that uses manually powered ratchets to index the platen and machine escapements to index the carriage.

CHAPTER 11

Clock and Watch Escapements

Clock and watch escapements are a special class of intermittent motion mechanism designed for a single task; to measure time. Although they are designed for just one purpose from an historical point of view, they are probably the most important intermittent motion mechanisms ever devised. Mechanical timekeepers made accurate navigation possible and so aided significantly the expansion of Western civilization. The measurement of time was also of pivotal importance to the development of science, and all time measurement until recently was accomplished by mechanical devices (or celestial observations). Timers were considered such an important tool of the scientist, in fact, that some of the greatest scientists of all time, men like Galileo; Huygens; Robert Hooke; and Wheatstone all devoted a portion of their professional lives to the improvement of clock and watch escapements.

We will be considering mechanical escapements primarily (with one or two electro-mechanical escapements thrown in for good measure) since our topic is intermittent motion mechanisms and not clocks and watches. The more recent electronic timers will not be considered at all, even though these are rapidly gaining in popularity and will probably soon replace mechanical devices in a great many situations. The cost of excellent mechanical clocks and watches has been reduced to a point that will be very hard for electronics to beat, however, thus mechanical timers will be with us for a long, long time to come.

Basic Requirements For a Mechanical Timekeeper

The basic requirement for any timekeeper is some phenomenon that can be used as a measure of time or "time base." The earliest clocks used the flow of water or sand through a fixed orifice; or the steady burning of a measured candle was used to track the passage of time. Sundials, of course, were also used. These basically analogue time bases were eventually replaced by systems in which a mechanical oscillator was used to measure the passage of time. Two basic mechanical oscillators quickly gained favor and are still in use today; the pendulum and the rotating-mass-spring oscillator.

A good mechanical oscillator has a cyclical motion which is relatively constant over a long period of time as long as energy lost to friction is replaced. In a clock or watch, the motion of the oscillator is used to regulate the velocity of a gear train which, in turn, drives hands or dials that indicate time. In effect, the clock mechanisms count the beats of the oscillating element (or integrate them) to convert "number of oscillations" to hours, minutes, seconds, etc. The clock or watch escapement is the mechanism that couples the rotating-spring-mass or pendulum to the gear train, and time indicators.

There are some severe restrictions on the escapement if it is to function correctly. It must accurately sense the motion of the mechanical oscillator (mass-spring. or pendulum) and use this information to control the motion of the gear train; which is con-

tinuously urged forward either by a large mainspring or falling weights. Since the escapement is generally controlled by the oscillator, it inevitably takes energy from the oscillator in order to function. If it takes out too much energy or takes it out at the wrong time, however, it will disturb the vibration of the oscillator and will affect the accuracy with which the oscillator measures time.

The escapement, furthermore, must also put energy back into the oscillator to keep the oscillator in motion. If this were not done, the oscillations would rapidly die down; thanks to viscous or coulomb friction, as in Figs. 3–16 or 3–18. The amount of energy introduced to the oscillator must be just about equal to that taken out by the escapement and by friction losses in the oscillator, or the oscillations will tend to increase in amplitude. This would destroy the accuracy of the time base (or would destroy the clock!). Thanks to the efforts of generations of superb designers, all of these things are accomplished easily by relatively simple and low-cost clock and watch escapements. We do not have room to go into all of the ramifications of the design of a good timekeeper, or even of a good escapement, but later on we will discuss some of the ways several of these things were accomplished, as we look at examples.

Advantages and Disadvantages of Clock and Watch Escapements

Since this is a special class of mechanism, it would be unfair to consider its advantages and disadvantages relative to the other devices we have been discussing. Needless to say, timer escapements have no peer when the job to be done is to keep time. On the other hand, they have few advantages compared to ratchets, Genevas, clutches, stepping motors, etc., when the job to be done is anything other than keeping time.

This is not to say, however, that they cannot be useful to the designer of intermittent motion mechanisms for machinery. As we will see in a later chapter, there are such things as machine escapements and they bear a strong family resemblance to clock and watch escapements, from which they were, no doubt, derived. Some of the forms and ideas developed for timekeeping might very well find application in heavy-duty machinery.

The first clocks as a matter of fact *were* heavy-duty machines, with escapements only a foot or more in diameter easily controlling weights weighing tons. Some of these designs gave excellent service for centuries, with only routine maintenance. Even small timer escapements are surprisingly rugged devices and should find more application in other instruments than they do at present. Mechanical delayed bomb fuses of World War II, for example, would withstand decelerations of 30,000 times the acceleration of gravity and still go on ticking accurately. Try dropping your high-speed business machine or machine tool with their Genevas, ratchets, stepping motors, etc., out of an airplane sometime and see if the machine will still function! Even though the timer escapement is a very small, lightweight device, its strength-to-weight ratio is very high, and it will stand a remarkable amount of abuse.

Another typical characteristic of good clock and watch escapements is very long life (see Fig. 5–2); almost zero wear in some applications. Disadvantages include the requirement for precision manufacturing techniques and correct lubrication, intolerance of dirt, etc., at least where precision timekeeping is required.

BASIC TYPES OF CLOCK AND WATCH ESCAPEMENTS

There are literally hundreds of different clock and watch escapements. Only a few of the more important ones will be considered in this text, however.

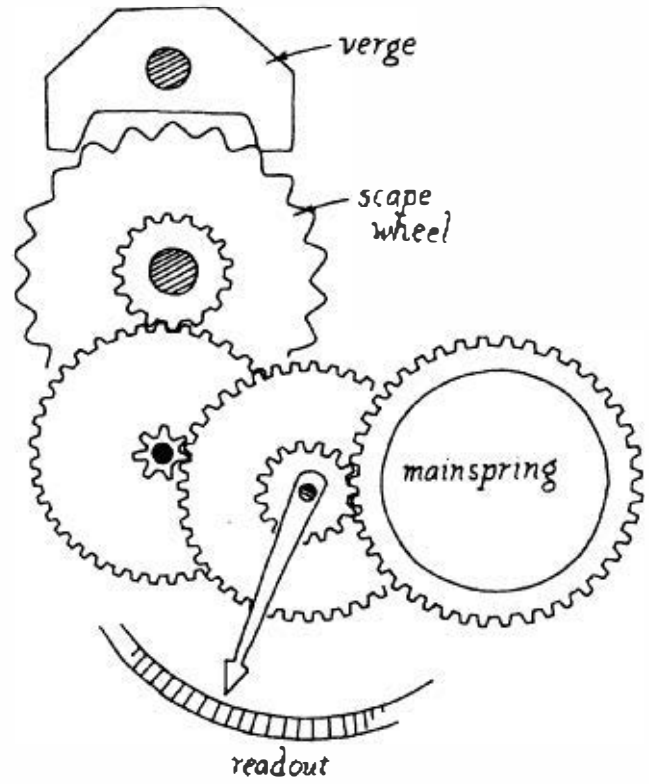


Fig. 11–1. Complete timer mechanism, showing drive spring, gear train and runaway, or verge, escapement.

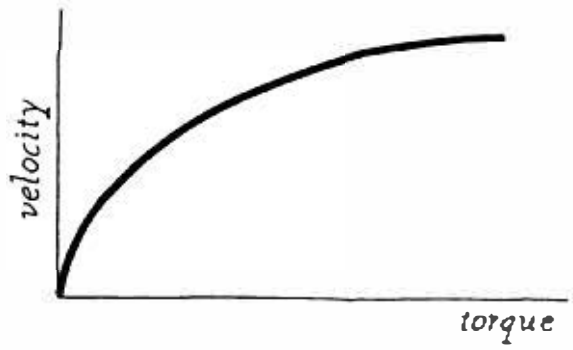


Fig. 11–2. Torque-speed curve for a runaway escapement.

Runaway Escapement

The simplest escapements use a non-resonant oscillating mass rather than an oscillating spring-mass as the basic timekeeper. A modern version of this type of escapement is shown in Fig. 11–1. The verge is the oscillating mass. It interferes with the rotation of the escape wheel as the latter attempts to rotate under the influence of a mainspring-driven gear train. One tooth or pallet of the verge is always positioned to interfere with the motion of the escape wheel or "scape" wheel, as it is sometimes known, so the latter is never allowed to rotate freely.

If the torque applied to the escape wheel is increased, the mass will oscillate more rapidly and the wheel will rotate more rapidly, since the verge is not a true oscillator. Hence the name "runaway" escapement for this type of mechanism. Figure 1 1–2 shows a typical torque-speed curve for a verge escapement. This type of escapement, incidentally, has a very high "beat rate." It sounds more like a bumblebee than a watch. The runaway finds a great deal of application in such things as portable range timers, parking meters, and in military products such as bomb fuses, rockets, and the like, because it is very rugged and economical. Furthermore, it is a self-starting escapement, which is necessary in many military situations. Also, its characteristics are similar to those of the fluid damper, and the runaway escapement is often used as a mechanical damper for speed control.

Pendulum Clock Escapement

Figure 11–3 shows the clockworks from a typical pendulum clock. The pendulum is hung from a knifeblade by a thin spring or strap, to reduce bearing friction effects. A yoke fastened to the escapement engages the pendulum near the top. As the pendulum oscillates, it moves the scape (escape) lever, which in turn releases the scape wheel. This latter is constantly urged to rotate by the gear train which is driven by the large falling weight. Further gears couple the weight to the output indicator; in this case, a single hour hand. The motion of the pendulum, then, controls the motion of the falling weight; which motion is indicated by the rotation of the hour hand. At some point during each cycle of the pendulum (the maximum velocity, or vertical, position is preferred) the scape wheel pushes on the scape lever which in turn, pushes on the pendulum, replacing any energy lost through bearing supports, air friction and the like. Different types of clock escapement accomplish this transfer of energy in different ways.

Watch Escapement

A typical watch mechanism is shown in Fig. 11–4. The pendulum has now been replaced by an oscillating springmass—the mass being called a "balance wheel," and the spring being called the "hairspring." The motion of this oscillator controls an escapement

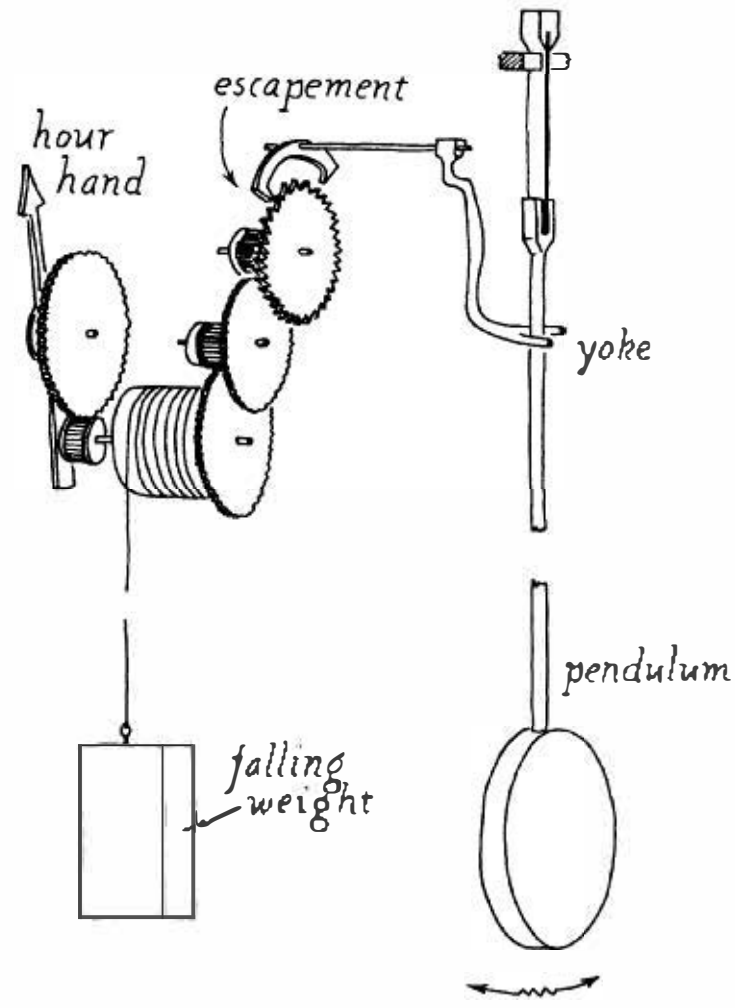


Fig. 11–3. Pendulum clock. Shows the relationship between mechanical oscillator (the pendulum); the clock escapement; the input power source (a falling weight); and the output indicator (hour hand).

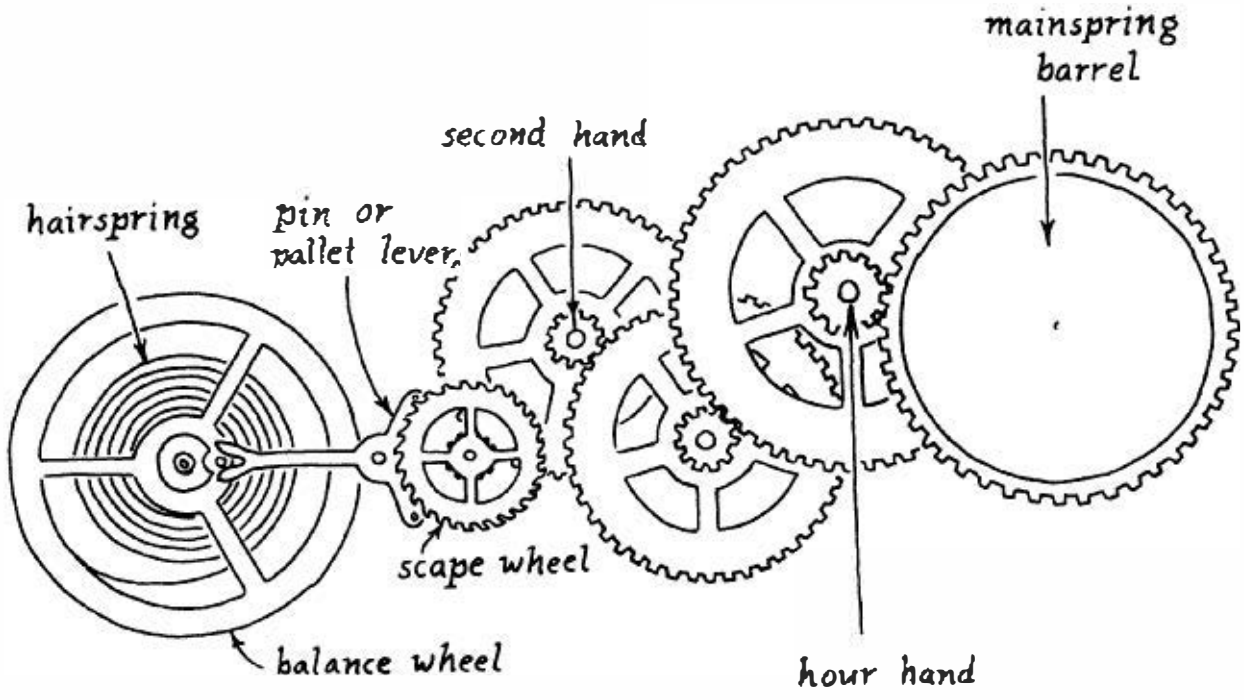


Fig. 11-4. Pocket watch showing the relationship between power source (mainspring), gear train, escapement, mechanical oscillator (balance wheel and hairspring) and indicators (second and hour hands). The minute hand is driven by additional gears (not shown) from the hourhand spindle.

which in turn regulates the motion of a gear train. In this case, however, the gear train is driven by a large mainspring rather than by a falling weight. Hour and second hands are in turn driven by the gear train, as shown in the illustration. The minute hand would be mounted on the same shaft as the hour hand but would be driven by different gears. I have omitted it for clarity. Again, energy must be delivered from the main power source, the mainspring, to the oscillating balance wheel, by the escapement, at some point during each cycle of oscillation.

And, again, it is best to introduce such energy at the maximum velocity point since an impulse on the balance wheel at this point will have the least effect on the frequency or period of oscillation. Different watch escapements accomplish this energy transfer in different ways. Watch escapements designed for maximum timekeeping accuracy (i.e., minimum interference with the oscillator) inherently tend to be weak in selfstarting, but the handling they receive when they are wound is generally sufficient to get them going.

Motion Curves

Figure 11-5 shows typical motion curves; angular acceleration, α; angular velocity, ω; and angular displacement, θ for the scape wheel (and, therefore for any element of the gear train) in a clock or a watch. This picture is also a reasonably correct description of the motion of the gear train in a runaway escapement, although, in this case the velocity ramps would appear "heel to toe" with essentially no sustained dwell periods.

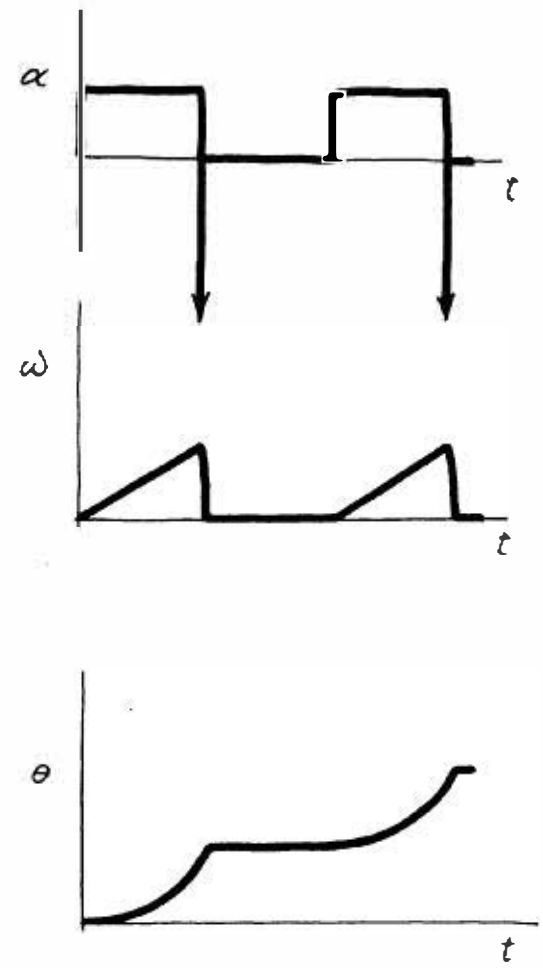


Fig. 11-5. Motion curves for a clock or watch escapement for one complete cycle of a pendulum or balance wheel (a tick and a tock).

As can be seen, a net torque (from the mainspring or falling weight) is suddenly applied to the scape wheel when the latter is released by the scape lever or verge. The wheel then moves until striking the other tooth (or pallet, as it is called) of the escapement; this suddenly stops the entire system. Motion recommences when the escapement permits. Two steps (a tick and a tock) are shown in Fig. 11–5.

Miscellaneous Problems

A mechanical clock or watch can be a superb timekeeper if sufficient attention is paid to design details. In addition to the problems already discussed, great care must be taken to be certain that the bearing friction in the balance wheel, scape wheel and scape lever shafts does not change significantly during the useful life of the device. In the best timepieces, furthermore, provision is made to compensate for temperature effects (which would cause oscillating masses to increase in size, springs to increase in length, etc.). For those clocks and watches subjected to shock, vibration, or moisture, etc., attention must be given to shock-loaded pivot bearings, careful selection of assembly techniques, sealed cases, etc. And, of course, cost is always a factor, with the designer striving for maximum performance at minimum cost. The extent to which cost reduction has been carried in the watch industry is really amazing.

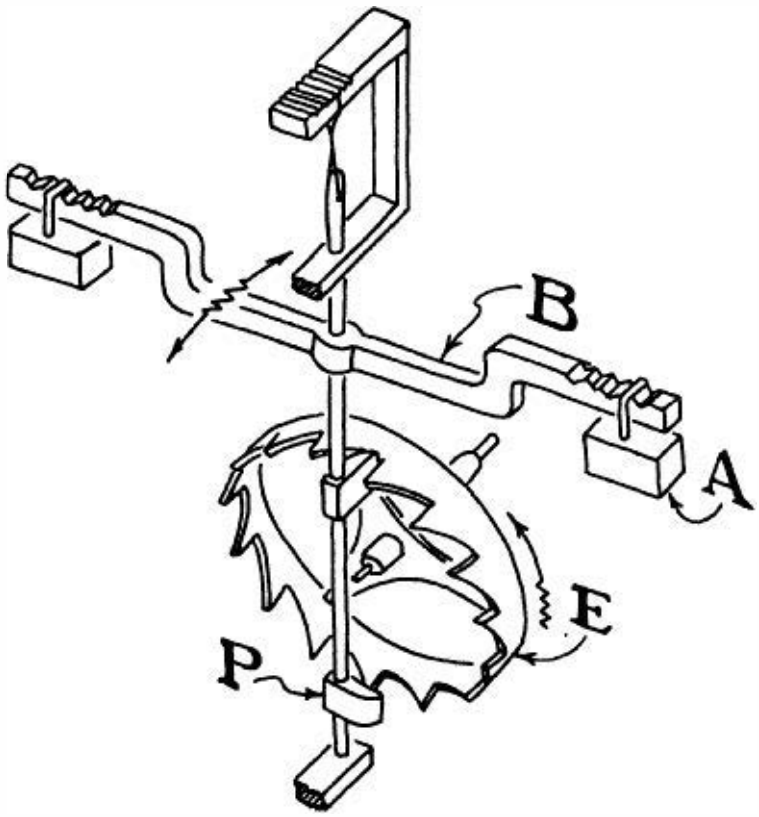


Fig. 11-6. Foliot verge runaway escapement.

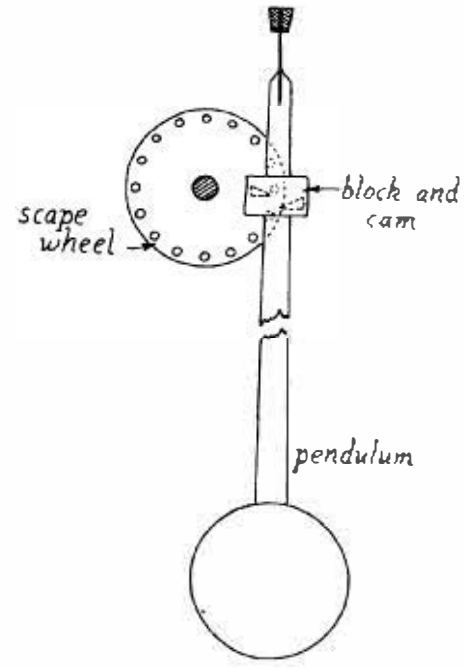


Fig. 11-7. Block and cam runaway escapement.

Varieties of Escapements

Let us look at some escapements. The first, the Foliot verge runaway in Fig. 11–6, is one of the oldest known. Probably the first successful clock escapement, it was widely used in steeple clocks for over three centuries (until the pendulum clock was invented about 1650 AD). This is a runaway escapement, there being only an oscillating mass (the arm *B*, and small weights *A*) to control the motion of the scape wheel (*E*), through the action of the pallets, *P*. There is no oscillating spring-mass combination. This device was, therefore, very torque sensitive; its speed would change if the applied torque changed; a serious problem in steeple clocks which are usually exposed to extremes of weather. Nevertheless, this was a very successful device in its day. The escapement used as a model for this drawing was about two feet high.

Figure 11–7 shows a block and cam runaway escapement, another ancient runaway escapement for a pendulum clock, nearly as old as that in Fig. 11–6, but far less popular. Pins (not teeth) on the scape wheel encounter small wedge cams on the block mounted on the pendulum. These impacts between pins and cams control the speed of the scape wheel, and deliver impulses to the pendulum to keep it swinging.

The small military timers shown in Fig. 11–8, left, are direct descendents of the early devices shown in Figs. 11–6 and 11–7, and use a modern verge escapement such as that shown at the bottom of the drawing at the right. This type of escapement is used in

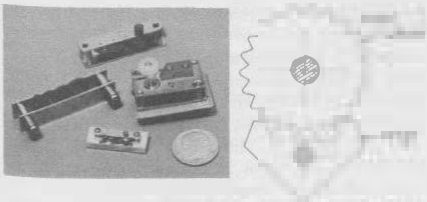


Fig. 11-8. Small military timers with modern verge escapements.

fuses, bombs, rocket systems, and so forth, because it is rugged, low-cost, self-starting, and reliable.

The Junghans cylinder escapement (Fig. 11-9) is another mechanism directly descended from those in Figs. 11-6 and 11-7. It has some of the characteristics of the gear-train-regulating-watch escapement. In Fig. 11-4, in that it, too, uses a [illegible] in conjunction with a [illegible] this [illegible], a balance arm. Both are, therefore, "tuned" to a specific beat frequency. Note that the [illegible]t of Fig. 11-4, detaches the balance mass from direct engagement with the escape lever, giving rise to greater accuracy. In effect, "detached lever" escapements represent an evolutionary refinement of the "tuned" cylinder escapement typified here. Escapements of the Junghans type provide considerably better accuracy than a runaway escapement, if not quite the accuracy of a good detached-lever mechanism. Like the verge, the Junghans escapement is rugged and economical [illegible] between those of a verge and the pocket watch.

The next seven illustrations, Figs. 11-10 through 11-16, show various types of pendulum clock escapements arranged roughly in the order of their antiquity. All of these mechanisms were good timekeepers with the last one in this group being the best single-pendulum escapement ever devised.

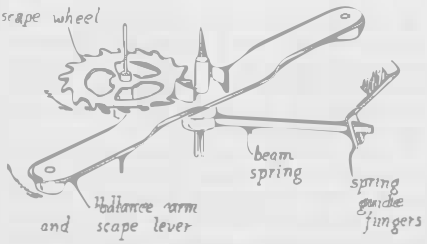
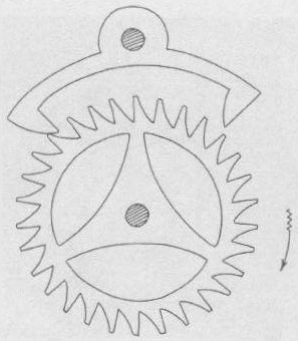
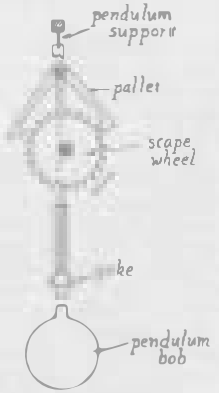


Fig 11-9. Junghans cylinder escapement.



A recoil escapement from a pendulum clock, shown in Fig. 11-10, was the earliest form of pendulum escapement. The scape wheel backs up each time an escape-lever tooth is extracted from the wheel. This caused some timing errors, but the recoil escapement was very popular in its time.

Figure 11-[illegible] shows a recoil clock escap[illegible]



Drawing courtesy of MACHINE DESIGN Magazine; Dec [illegible] p. 181 ff

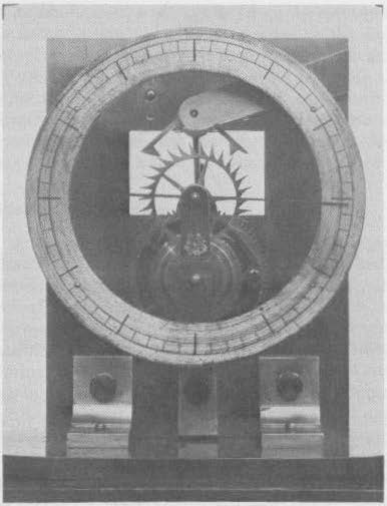


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Fig. 11–12. Working model of an "anchor" recoil escapement.

the relationship between the escapement and the oscillating pendulum.

A working model of a recoil escapement is shown in Fig. 11–12. This type of escapement is often called an "anchor" escapement. Historians believe it was invented by Robert Hooke around the end of the 17th century.

Figure 11–13 illustrates the deadbeat clock escapement invented by George Graham, around 1715. This is an improved pendulum escapement in which the scape wheel does not back up as the escape-lever teeth are extracted. The device resulted in a considerable improvement in accuracy over that obtained with a recoil escapement.

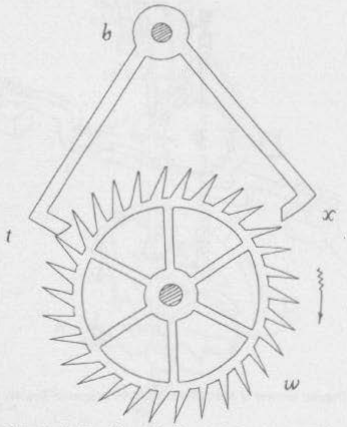

Fig. 11–13. Deadbeat clock escapement, ca. 1715.

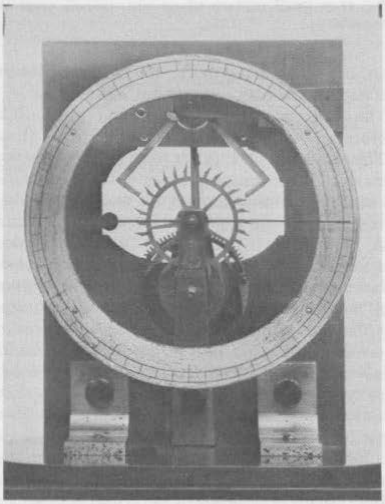

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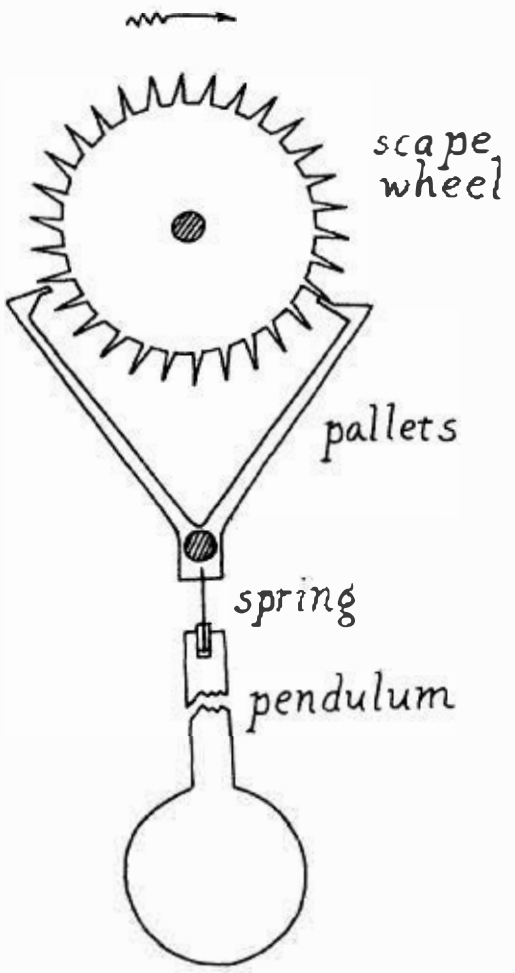
Fig. 11–14. Working model of Graham's deadbeat escapement.

A working model of a deadbeat escapement is shown in Fig. 11–14. This device was one of Graham's many contributions to the design of clocks and watches. His contributions were considered to be so important that he was honored by his nation with burial in Westminster Abbey.

Not all "clockworks" are as small and "delicate" as those we usually encounter, as seen in Fig. 11–15, which is Lord Grimthorpe's three-legged gravity escapement, invented in 1854, and used in the Big Ben of that time. The gravity arms are lifted by three pins on the scape "wheel" and are subsequently

released by the pendulum. This controls the motion of the three-legged scape wheel which periodically impacts small stops or blocks on the gravity arms. The scape wheel, of course, is urged to rotate by a gear train, which, in turn, is powered by falling weights as in other pendulum clocks. A great advantage of this escapement is that all the impulses to the pendulum are derived from the gravity arms, rather than from the scape wheel, through a yoke as in the recoil or deadbeat escapements. This means that the impulses to the pendulum are constant even if the load on the gear train varies, as it usually does in turret clocks that are exposed to the weather. The gravity arms, incidentally, are about one-foot long on this design; to give you some idea of the size of the escapement.

Riefler's escapement, shown in Fig. 11–16, is considered to be the best, purely mechanical pendulum escapement ever designed, because the impulses are imparted to the pendulum through a supporting spring only (by flexing the spring as the pallet lever oscillates). The pendulum is otherwise completely free and is, therefore, a superb mechanical oscillator. The only pendulum clocks which have been found



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 11–16. Riefler's escapement.

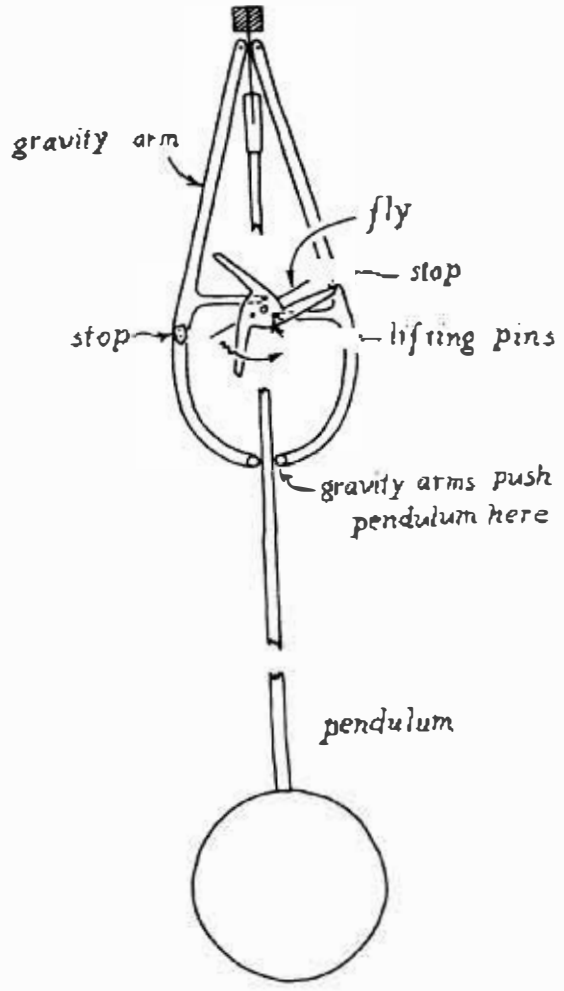
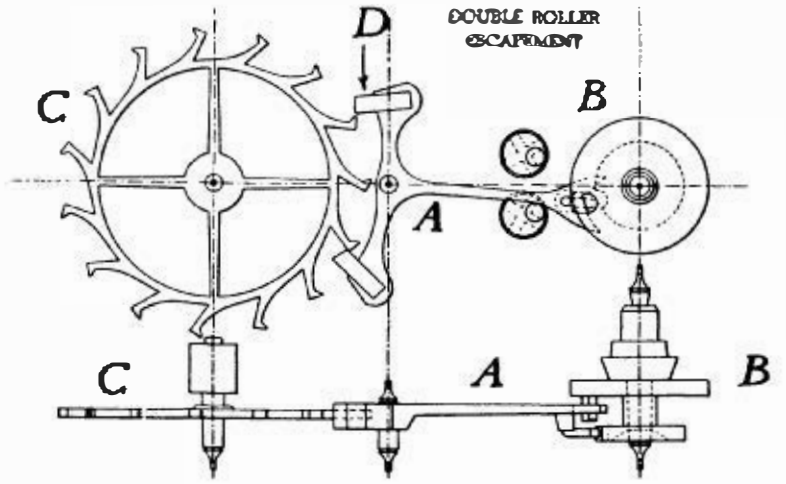


Fig. 11–15. Lord Grimthorpe's three-legged gravity escapement, ca. 1854.

to be more accurate than this use electro-mechanical escapements with slave and master pendulum systems.

Following are six illustrations showing various types of watch escapements. The main difference between clocks and watches is that the latter are portable. The former usually use pendulum oscillators to keep time, while the latter rely on balance wheel-spring combinations.

This detached pallet lever watch escapement (Fig. 11–17) is one of the most accurate and popular of all time. It is called a detached lever escapement because, like the typical watch escapement of Fig. 11-4, the scape wheel and balance wheel (oscillating mass) are connected by a lever that does not touch the balance wheel at all except for an instant in the middle of each oscillation (maximum velocity point), leaving the mechanical oscillator free during most of each cycle. This results in a great improvement in timing accuracy. Impulses to the balance wheel are delivered by the detached lever, which, in turn, is impulsed by the scape wheel at the end of each

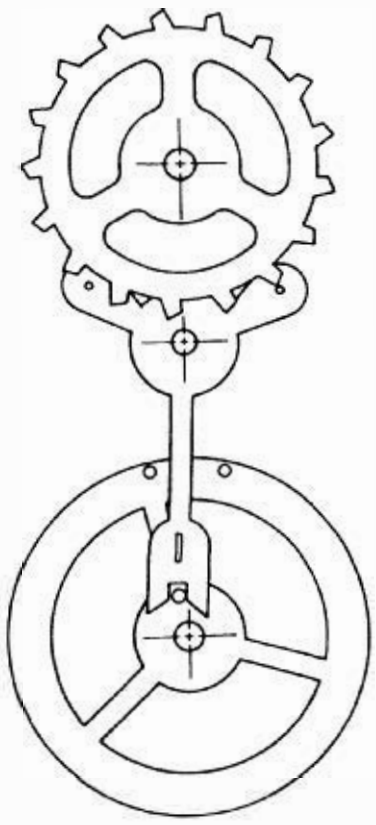


Courtesy of the Waltham Precision Instrument Company

Fig. 11–17. Detached pallet lever watch escapement.

motion. The escapement is not self starting, although the slightest motion in handling will get it going. The following elements can be identified in the illustration: *A*. Detached lever; *B*. Balance wheel roller (mounted under the balance wheel—not shown—and on the same shaft); *C*. Scape wheel; *D*. Pallets.

This detached pin lever escapement, Fig. 11–18, is very similar to the one shown in Fig. 11–17, except that the pallets on the detached lever have been replaced by small pins. Since pallets are usually

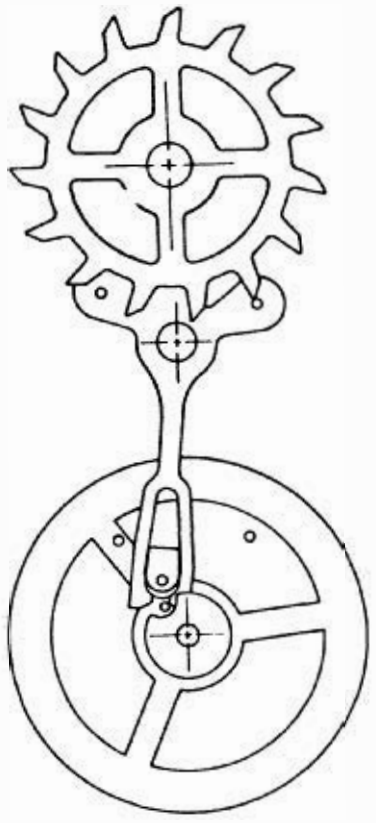


Courtesy of the Raymond Engineering Company

Fig. 11–18. Detached pin lever escapement.

jewels, the pin lever escapement is a much cheaper proposition. It is used in low-cost pocket and wrist watches, but is an excellent timekeeper; one that is cheap, rugged, and accurate.

In this semi-detached pin lever escapement, Fig. 11–19, the lever contacts the balance wheel at all times. The advantage of this is that the escapement is more nearly self-starting than is a fully detached mechanism which has only a weak tendency to self start. A semi-detached escapement, therefore, finds applications in military devices where reliability is



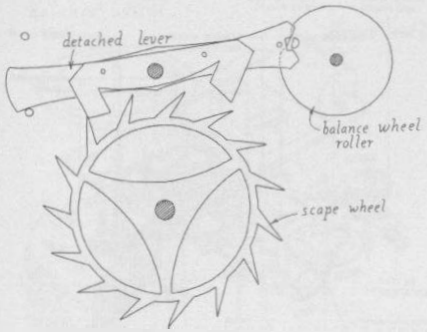
Courtesy of the Raymond Engineering Company

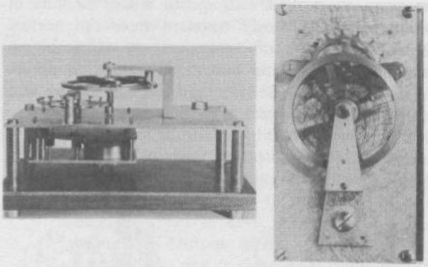
Fig. 11–19. Semidetached pin lever escapement.

essential and the device may not be handled (as is a pocket or wrist watch) to start it.

Schematic of an English lever escapement (*Top*, Fig. 11–20), invented by Thomas Mudge, about 1755. This is another detached lever design, not quite as successful as the ones shown in Figs. 11–17 or 11–18, but included here to show that a bewildering variety of escapements has been developed. The photograph is of a large working model of a similar Swiss lever escapement.

Figure 11–21 shows the Tavaro cylinder escape ment. The scape wheel contacts a cylindrical and partially hollow shaft on the balance arm in this unusual design which incorporates a balance-arm suspension via a torsional hair spring located on the



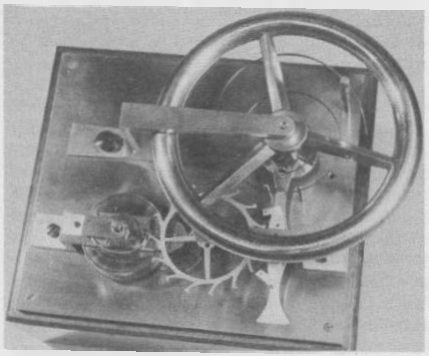


British Crown copyright, Science Museum, London

Fig. 11-22. Large-scale model of a cylinder escapement. (*Left*) Side view; (*Right*) Top view.

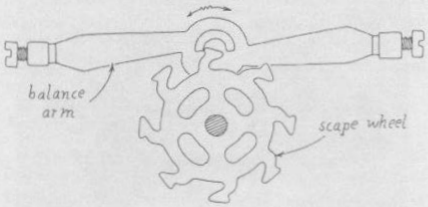
axis of the cylinder. Although the scape wheel bears on the balance arm almost continuously, its influence on the balance arm is minimized by the fact that the contact is made at a very sma[illegible] hence, low torque). The [illegible] is not a common escapement, but a relatively successful one for military projectile use where spin would adversely affect the more common spiral hairspring. (Note [illegible] es to the Junghans escapement of Fig. 11-9.)

Side, and top view photographs of a large-scale working model of a cylinder escapement are shown in Fig. 11-22.



Photograph, British Crown copyright, Science Museum, London

Fig. 11-20 [illegible] ment, ca. 1[illegible] (*Bottom*) Working model of a Swiss lever escapement.



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965: p. 121 ff

Fig. 11-21. Tavaro cylinder escapement.

Chronometers are spring-powered watches that are designed specifically for marine navigational use. They are the most precise watches ever built, and as early as 1759 were able to [illegible] with errors of only 1.5 seconds in five months. This made it possible for an explorer to arrive on the far [illegible] of the ocean on a dark and stormy night and fin[illegible], as planned, at the mouth of a large, navigable river or bay instead of on one of the rocks along the coast. Chronometers have temperature-compensated bal-

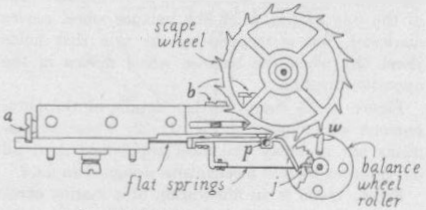
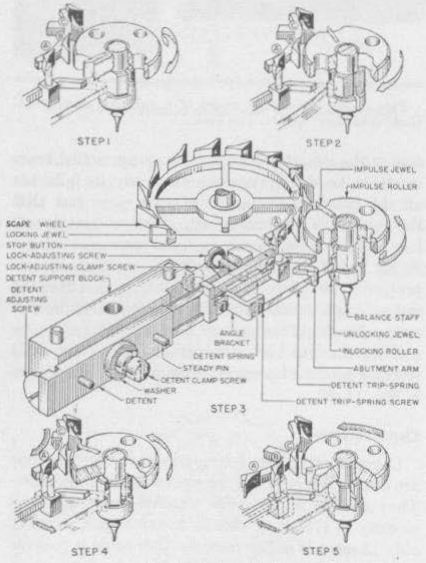


Fig. 11-23. Chronometer escapement.

ance wheels; Elinvar hair springs whose modulus of elasticity stays nearly constant over the normal operating temperature range, and are adjustable on almost every [illegible] Part. In the modern design shown in Fig. 11–23, for example, screws (*a*) and (*b*) control the position of the single "pallet" or locking jewel (*p*). The balance wheel in this design receives periodic impulses through jewel (*w*). The locking pin (*p*) is periodically forced to release the



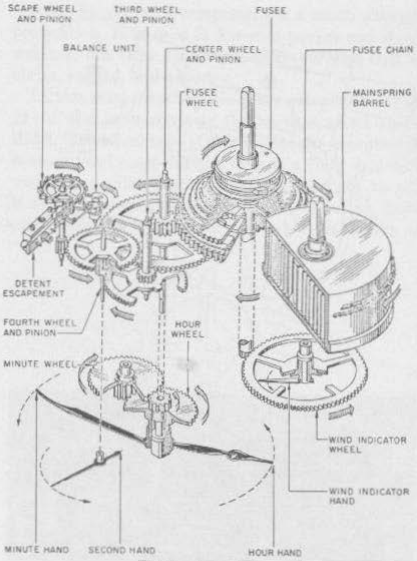
Drawing courtesy of the Hamilton Watch Company

Fig. 11–24. Details of the chronometer escapement shown in Fig. 11–23.

scape wheel by the action of the unlocking jewel [illegible] a flat spring [illegible] is free to [illegible] of the [illegible] (*j*) when the balance [illegible] moves clockwise, but is trapped by the arm that holds jewel (*p*) when the balance wheel moves in the [illegible]

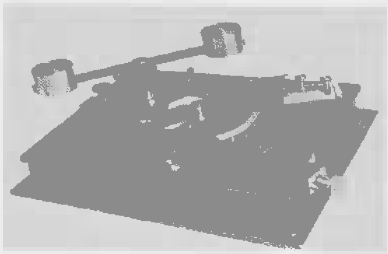
Figure 11–24 shows further details of the chronometer escapement of Fig. 11–23. These mechanisms are probably the most sophisticated of all [illegible] motion mechanisms designed to date.

Figure 11–25 is an illustration of a [illegible] chronometer, showing the relationship between the chro-



nometer escapement, the power source ([illegible]ing) gear[illegible], and [illegible] (second, rai[illegible], and hour hands. [illegible] with Fig. 11–4 shows that both pocket-watch and chronometer escapements are essentially the same. They only [illegible]

A large-scale working model of a marine chro-



British Crown copyright, Science Museum, London

Fig. 11–26. Working model of a marine chronometer escape-

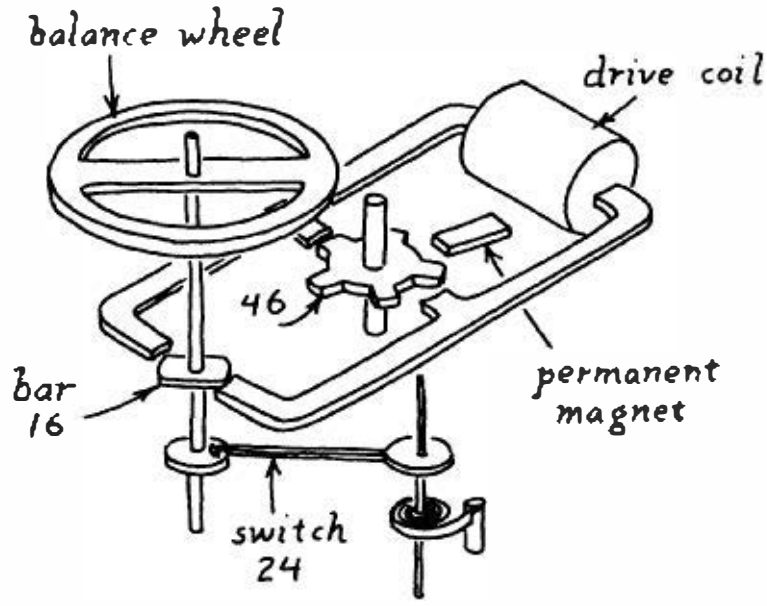


Fig. 11-27. An electro mechanical watch. (U.S. Patent 3,147,583; D. Fritsch.)

nometer escapement in The Science Museum of London, is shown in Fig. 11-26. This is an old form of chronometer escapement but it has most of the features of the modern escapement shown in Figs. 11-23 and 11-24.

Electro mechanical clocks and watches have been gaining popularity in recent years, thanks in part to the availability of small, efficient batteries. Also, designers have tried to replace tooth-to-tooth contact with magnetic forces to reduce friction in watch escapements. In the design (not necessarily typical) in Fig. 11-27 are shown the various elements required in an electrical-mechanical timekeeper. Magnetically transmitted impulses through *bar 16* keep the balance wheel oscillating. Switch *24* on the balance wheel will time these impulses, and also time the magnetic forces that drive (not escape) the "scape wheel," *46*. This latter is really part of a tiny variable reluctance stepping motor (see Chapter 14). Since considerable energy magnification (the ratio of the work required to close the switch, to the work available from the solenoid) is available here, it is not necessary to provide the mechanical advantage of an escapement; hence, the stepping motor approach. This is true of many electromechanical watch "escapements."

Anatomy of a Watch Escapement

Let us now take a closer look at some of the design features (Fig. 11-28) of one of the best mechanical timekeepers ever invented, the detached pallet lever escapement:

A. Scape wheel, output—loaded by gear train and mainspring. It tries to turn continuously, but is prevented from doing so by the scape lever. The teeth on the scape wheel are very carefully shaped by generations of science and experience. They must not lock on the scape lever, yet must be held easily by the lever until the next step is desired. Furthermore, when escaping, these teeth must provide a tiny bit of energy to the lever (which energy is derived originally from the mainspring) to be passed to the balance wheel to keep that member oscillating; B. Output pinion—connects scape wheel to the rest of the gear train; C. Scape lever—in this case, a "detached" lever because it is not normally engaged by the balance wheel roller. This lever normally

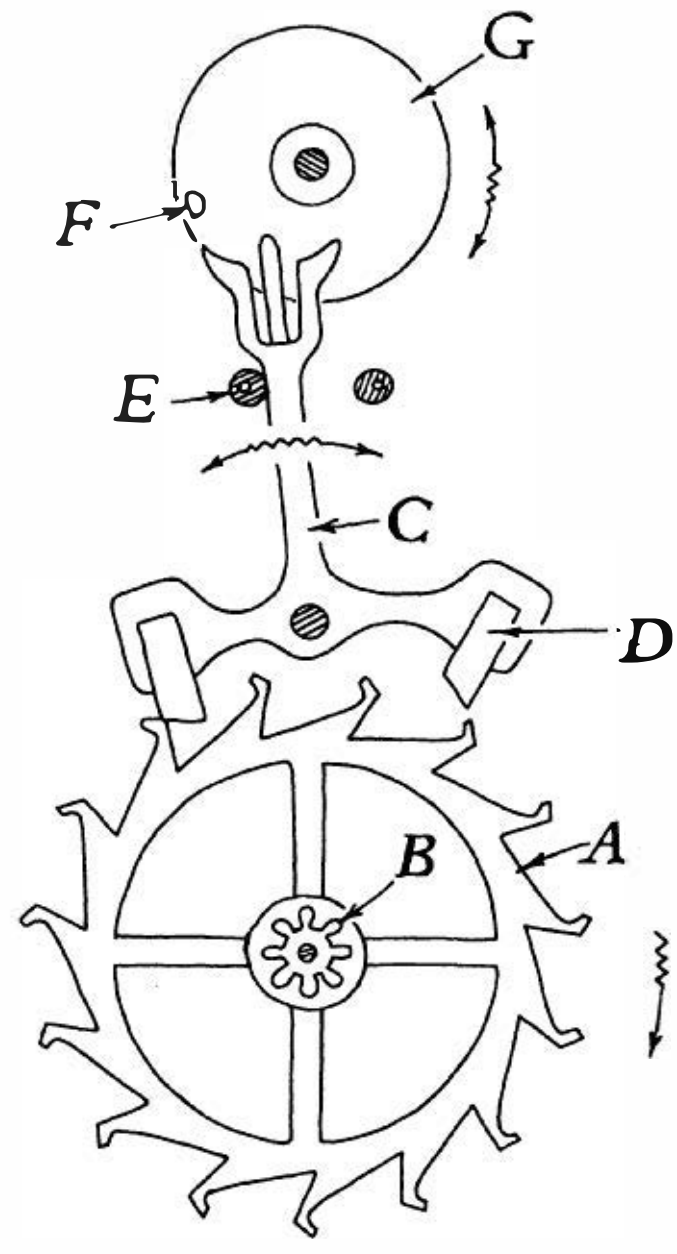


Fig. 11-28. Anatomy of a watch escapement.

rests against one of the two stop pins *E*, held (gently) by forces exerted on the lever, by the scape wheel *A*; D. Pallet—scape lever tooth—must be extremely hard, to operate with low friction for many years without benefit of lubrication (which would cause the pallets to stick to the scape wheel). In an expensive watch these are made of jewels and must be shaped to function properly together with the teeth of the scape wheel; E. Stop pins or banking pins—control length of throw of the scape lever (often eccentric for adjustment); F. Jewel impulse pin—provides the only contact between the balance-wheel assembly and the rest of the escapement. Periodically, it drives the lever from one stop pin to the other. As with the pallets, this pin must work a long time without lubrication; G. Balance-wheel roller—small cam mounted on the same shaft as the larger balance wheel. The latter, with the hairspring, provides the "time base" on which the watch depends. It must not touch the lever except for a brief period at the center (maximum velocity point) of each swing. Although it is the balance-wheel roller which drives the lever back and forth, the lever succeeds in giving the roller a brief push during part of each swing to keep the mechanical oscillator going.

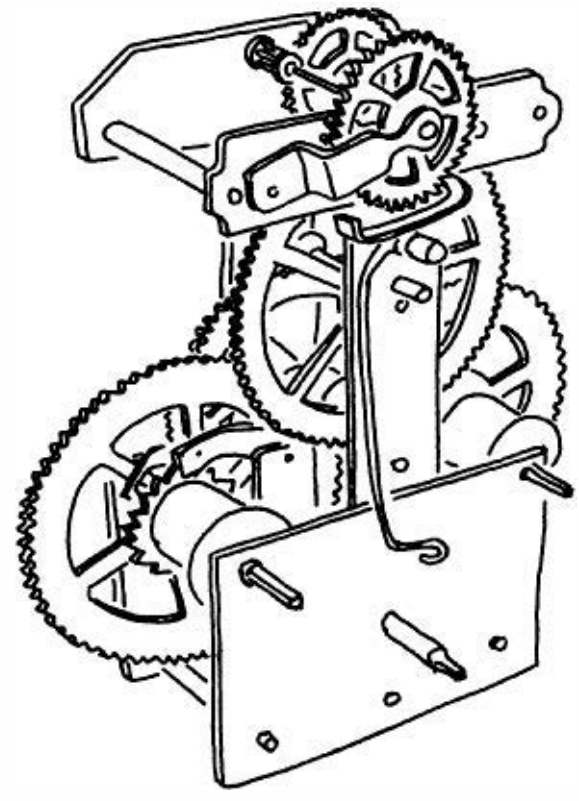


Fig. 11-29. Clock movement manufactured in Connecticut by Joseph Ives, ca. 1847.

Historical Note

As has been pointed out, the intermittent motion mechanisms developed for the purpose of keeping time have had an immeasurable influence on the development of our civilization. Of course, this was said in considering their use as instruments and tools. However, less obvious but also very real, is the influence these devices have had on manufacturing practices throughout the world. Many precision manufacturing techniques were first developed for the production of clocks or watches, where precision is essential to performance. Furthermore, these products were among the first to be built by mass production techniques, which is ample evidence of their popularity and importance. The sketch, Fig. 11-29, shows a clock movement manufactured by Joseph Ives in Connecticut, about 1847.

CHAPTER 12

Machine and Inverse Escapements

MACHINE AND INVERSE ESCAPEMENTS IN GENERAL

In this chapter two more types of mechanisms will be considered—machine escapements and inverse escapements. Neither of these is as popular with machine designers as are ratchets, cams, or Genevas, but both deserve more attention than they get, since they offer the designer significant advantages in many design situations. Both machine and inverse escapements look very much like clock escapements, but neither is designed to run synchronously and both are designed to handle considerably larger loads than the typical clock escapement. Despite the geometrical similarities between these two types of mechanism, they will be discussed separately (machine escapements first) because they differ quite a bit in their operation.

MACHINE ESCAPEMENTS

Description of Machine Escapements

A typical machine escapement is shown in Fig. 12–1. As can be seen from the drawing it looks like a clock escapement except that the scape lever is controlled by the rotation of a snail cam rather than by a pendulum. The scape lever, of course, could also be controlled by a solenoid rather than by an input cam. This would place the length of the dwell period under the control of the operator. In either case, one of the two teeth (pallets) of the scape lever is always in a position to interfere with the motion of the scape wheel which, therefore, can never run away.

As in a clock escapement, the scape wheel in Fig. 12–1 is continuously urged to rotate by the machine to which it is fastened. A spring or weight could be used, as in a clock or watch, but a slip clutch or stallable motor is more commonly used in machine design. Sometimes, both a spring and slip clutch are used, the clutch being used to limit the torque that keeps the spring wound. Spring loaded escapements are usually called "load-and-fire" escapements. Examples of these will be seen later on.

The machine escapement is similar in some ways to certain types of limited motion clutches. In the next chapter a one-turn ratchet clutch will be seen, for example, that might well be mistaken for a machine escapement. The difference is that an escapement controls or releases energy in some other mechanism; a clutch transmits energy from drive to load.

Figure 12–2 shows another machine escapement. It differs from Fig. 12–1 in that the scape lever has only a single pallet. This mechanism will function correctly only if the scape lever moves rapidly enough to catch each tooth of the scape wheel after every motion of the wheel. As mentioned in Fig 12–1 one of the two pallets of that scape lever is always in a position to interfere with the motion of the scape wheel. The mechanism of Fig. 12–2, relying as it does on a race between the scape lever and the scape wheel, is a more risky mechanism; but, never-

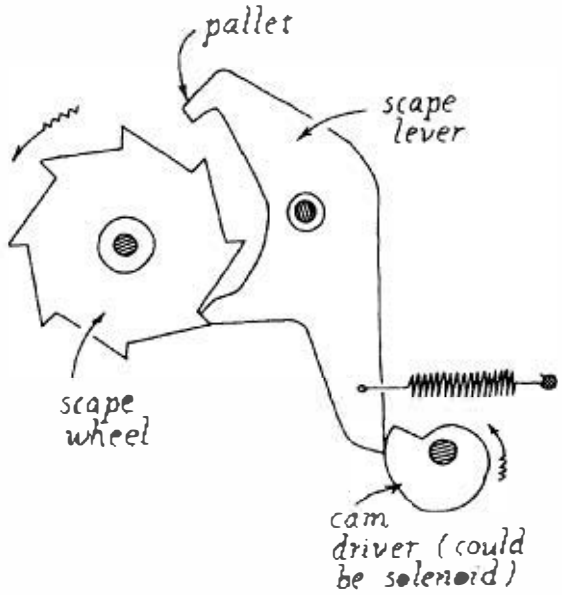


Fig. 12–1. Typical machine escapement.

theless, the single pallet escapement can be made to function correctly with proper design.

One way to help the single-pallet scape lever function correctly is to spring load it in the manner shown in Fig. 12–3A. Every time the pallet escapes a tooth on the wheel the lever, provided with a slot, immediately jumps ahead and downward (Fig. 12–3B). This trick gives the lever more time to re-engage the wheel (or catch the next tooth). The spring also acts to cushion the impacts that occur when the wheel is stopped by the lever.

Advantages and Disadvantages of Machine Escapements

One favorable attribute of the machine escapement is the fact that it has a large mechanical advantage. A small amount of control energy from a low torque rotating shaft or a lightweight solenoid can control (release) a large amount of energy or torque in the scape wheel and remainder of the machine. The only energy which has to be supplied to the control system is that required to extract the scape lever from the scape wheel. The ratio between this energy or torque and that which is controlled, is the same as the ratio between the normal force on the lever and the frictional resistance to its motion. Since the friction force can usually be kept at 10 or 20 percent of the normal force, one unit of energy can easily control ten. With special attention paid to the shape and surface treatment of the scape lever and wheel teeth, with proper balancing of scape lever, etc., the control to output torque ratio can be even greater than this.

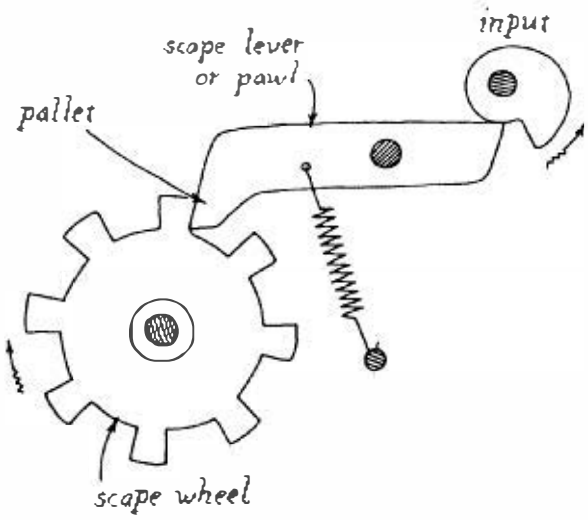


Fig. 12–2. Machine escapement with single-pallet scape lever.

This mechanical advantage leads to an additional advantage. The machine escapement can be operated

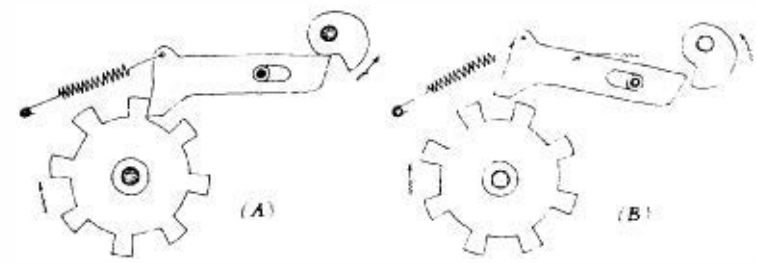
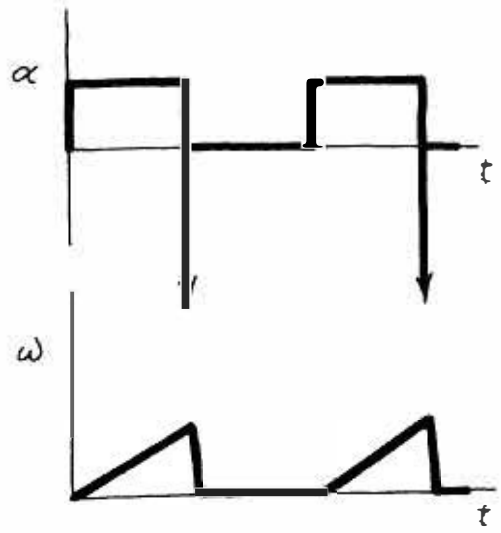


Fig. 12–3. Single-pallet machine escapement. (A) Engaged. (B) Disengaged.

at very high speeds because we do not have to wait for large drive torques to be developed by the system. Such torques are always present (produced by stalled motors, wound springs, slip clutches, etc.) urging the scape wheel forward. The escapement need only produce a small control torque which releases the drive torque. Solenoids, motors, or springs which drive the scape lever, therefore, can be small and the inertia of the escapement parts can be very small. All of this can result in very high speeds (with proper design of the drive circuits, etc.).

A different advantage can be obtained with the "load-and-fire" type of machine escapement. Here, a low-power motor can be used (with suitable gearing) to wind a large spring slowly. A very small control solenoid or equivalent is then used to actuate an escapement to release the energy in this spring. The system produces large output torques with low input energy and torque requirements. Of course, cycle rates have to be slow enough to allow the low-energy motor to wind up the high-torque spring, but this is not a serious disadvantage in many cases.

Another advantage is the machine escapement; it is a very simple device, even simpler than a ratchet. There is no need for such things as a no-back pawl, non-overthrow pawl, pawl springs, etc. Simplicity, in turn, leads generally to lower costs and to improved reliability. Finally, the machine escapement keeps the load under control quite reliably, although there is a period during which the load is free-wheeling between stops. If the escapement is designed properly, the load (scape wheel) cannot get very far before it is stopped (motion control).



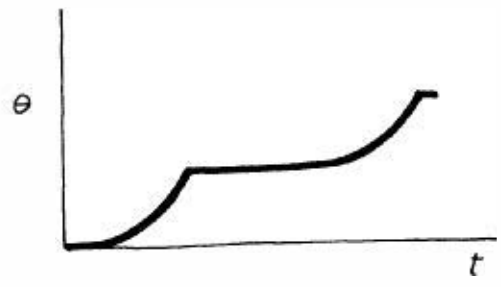


Fig. 12–4. Motion curves of a typical machine escapement. Notice the similarity to the curves of a clock escapement in Fig. 11–5.

There are disadvantages to machine escapements, however. This is an impact mechanism, probably producing more severe impacts than a ratchet, since full drive torque is usually applied to the scape wheel at all times. This means that machine escapements can only be used in relatively light-duty applications and can only take short steps (perhaps one revolution, as a maximum) to reduce impacts. Motion times are always short, therefore, but the dwell period can be placed under the control of the operator, if this is desirable.

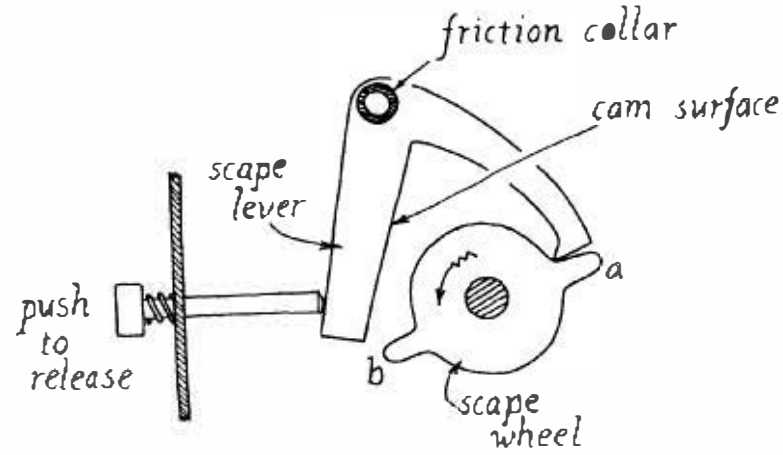


Fig. 12–5. Manually operated escapement in a towel dispenser.

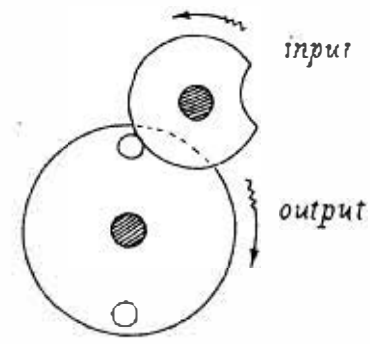
Motion Curves—Machine Escapements

Motion curves for the scape wheel of a typical machine escapement are shown in Fig. 12–4. These are identical to the motion curves for a clock escapement and are characterized by a constant torque period (following release of the scape wheel) followed by a severe impact as the wheel is stopped when it strikes the next pallet of the scape lever.

Examples of Machine Escapements

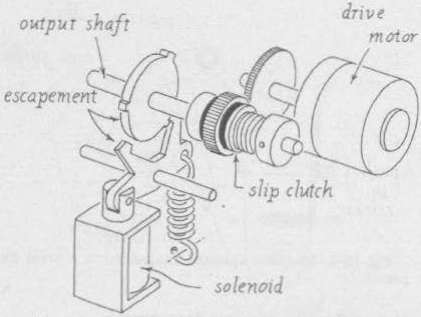
In this manually operated escapement, from a towel dispenser (Fig. 12–5), the operator pushes the control button which moves the scape lever counterclockwise until it strikes the wheel. He is now free to pull a towel from the dispenser. This act rotates the scape wheel, and tooth *a* of the scape wheel acts on the cam surface of the scape lever to move the scape lever back into the position shown in the illustration. This brings the right-hand pallet of the scape lever into position to interfere with tooth *b* on the scape wheel, and thus prevent the operator from getting more than one towel.

Figure 12–6 illustrates a very simple machine



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 181 ff

Fig. 12–6. Simple machine escapement with a rotating input.

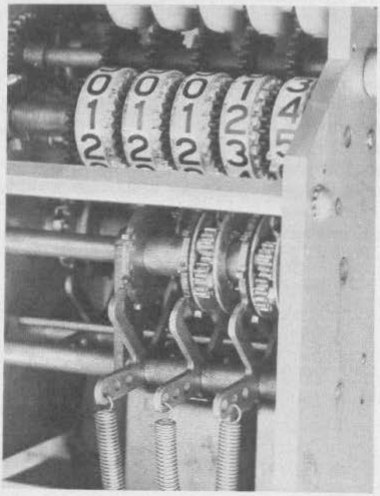


Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 12–7. Complete machine escapement system.

escapement with a rotating input. However, to function satisfactorily, the speed of the input must be faster than that of the output.

A complete machine escapement system is shown in Fig. 12–7. A drive motor operating through a slip clutch applies torque continuously to an output shaft. This remains at rest, however, until the control solenoid drives the escapement to release the output shaft briefly. This system is shown in Fig. 12–8, incorporated in a high-speed industrial counter. The speed of this counter is much higher than that which could be obtained reliably from a low-cost ratchet, inverse escapement, etc.



Photograph courtesy of Veeder-Root Company

Fig. 12–8. High-speed industrial counters driven by an escapement system.

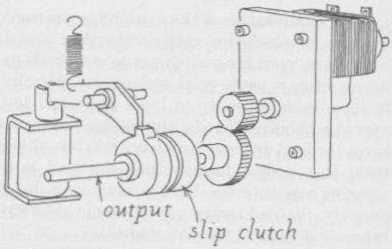


Fig. 12–9. Single-pallet machine escapement.

In a single-pallet machine escapement (Fig. 12–9), the output shaft will rotate one revolution every time the solenoid is actuated. The scape lever has only a single pallet, therefore the solenoid can only be actuated momentarily if the pallet is to re-engage the scape wheel in time to catch it at the end of the first revolution.

The device shown in Fig. 12–10 is a solenoid actuated escapement used to control the motion of a

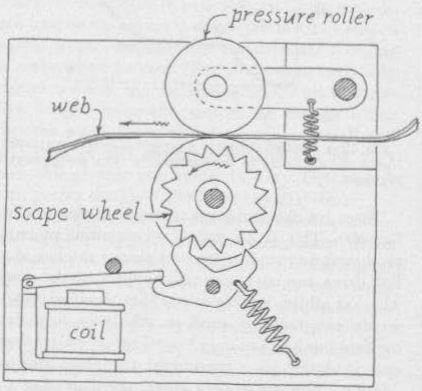
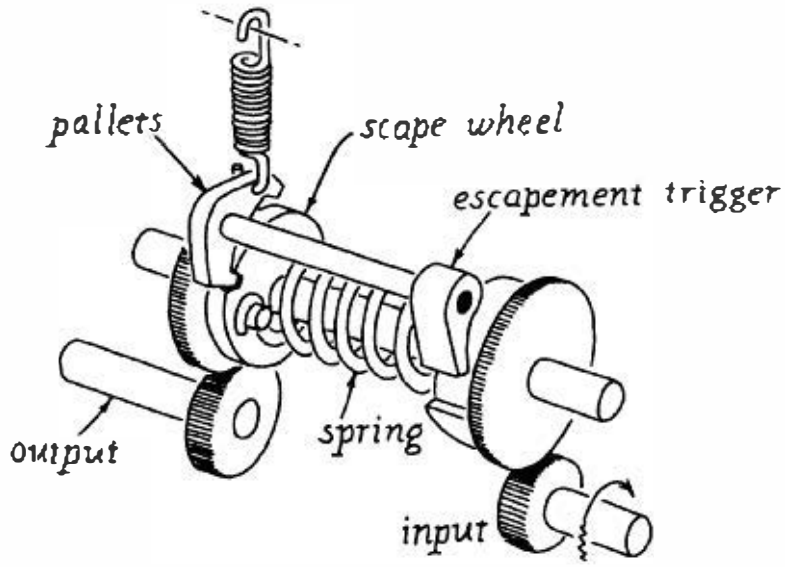


Fig. 12–10. Solenoid actuated escapement for controlling the motion of a web or belt. (U.S. Patent 3,504,558; W. J. Zenner.)



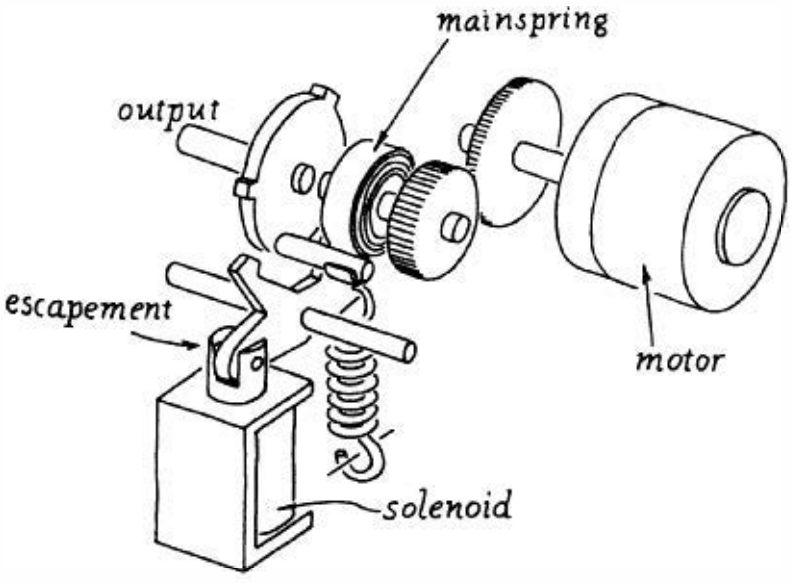
Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965: p. 121 ff

Fig. 12–11. Load-and-fire escapement.

web or belt. The scape wheel could be powered by a stalled motor (not shown).

The input shaft of this load-and-fire escapement (Fig. 12–11), rotates continuously. The spring is wound until the escapement trigger on the right-hand side is driven outwards by a cam fastened to the large gear on the right. This action moves the scape lever, allowing the scape wheel to be driven by the spring. Both dwell and cycle times are controlled by the mechanism.

Figure 12–12 shows another load-and-fire escapement. In this design, a clock spring is used to increase the amount of energy which can be stored in the system between index cycles. The motor should be a stallable motor in the arrangement shown, or a



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965: p. 121 ff

Fig. 12–12. Load-and-fire escapement with clock spring for storing energy between index cycles.

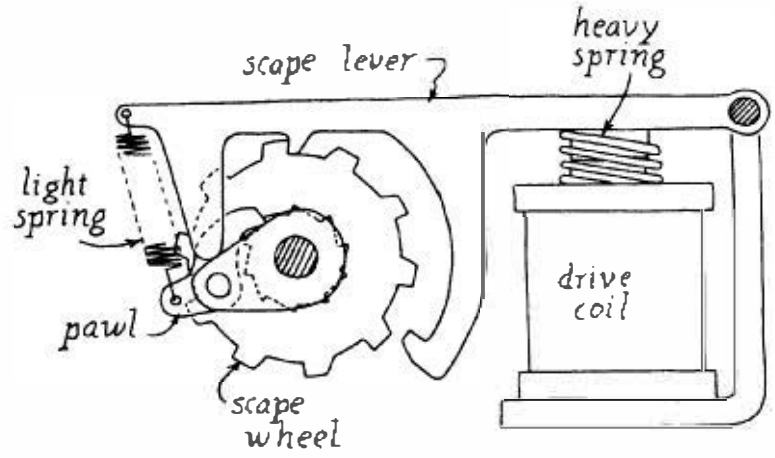


Fig.12–13 Solenoid powered load-and-fire escapement.

switch of some sort should be provided to fire the escapement prior to the end-of-wind of the main spring.

In the load-and-fire escapement of Fig. 12–13, the input energy is derived from a solenoid which loads a heavy spring. When the coil is de-energized the heavy spring moves the scape lever upward, loading the light spring and then releasing the scape wheel. The heavy spring drives the light spring, which drives the pawl, which drives the scape wheel—until this latter is caught by the lower scape lever pallet. The scape wheel does not rotate further when the drive coil is re-energized. (There should also be a no-back pawl since the input here is through a ratchet mechanism.)

The input of the low-cost load-and-fire escapement in Fig. 12–14 winds the spring of the right-hand shaft until the mutilated cylinder on the left-hand shaft releases the long scape lever (which is mounted on the right-hand shaft for bearing purposes, but which is not fastened to that shaft). This is actually

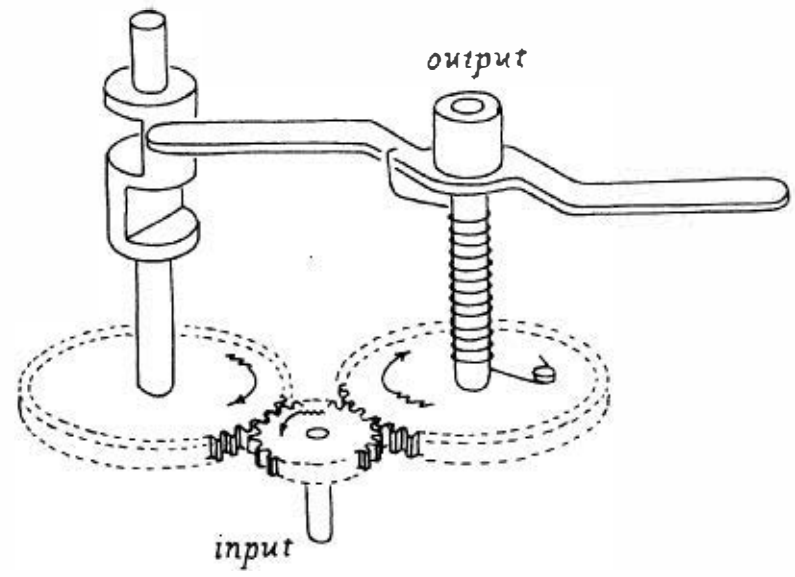
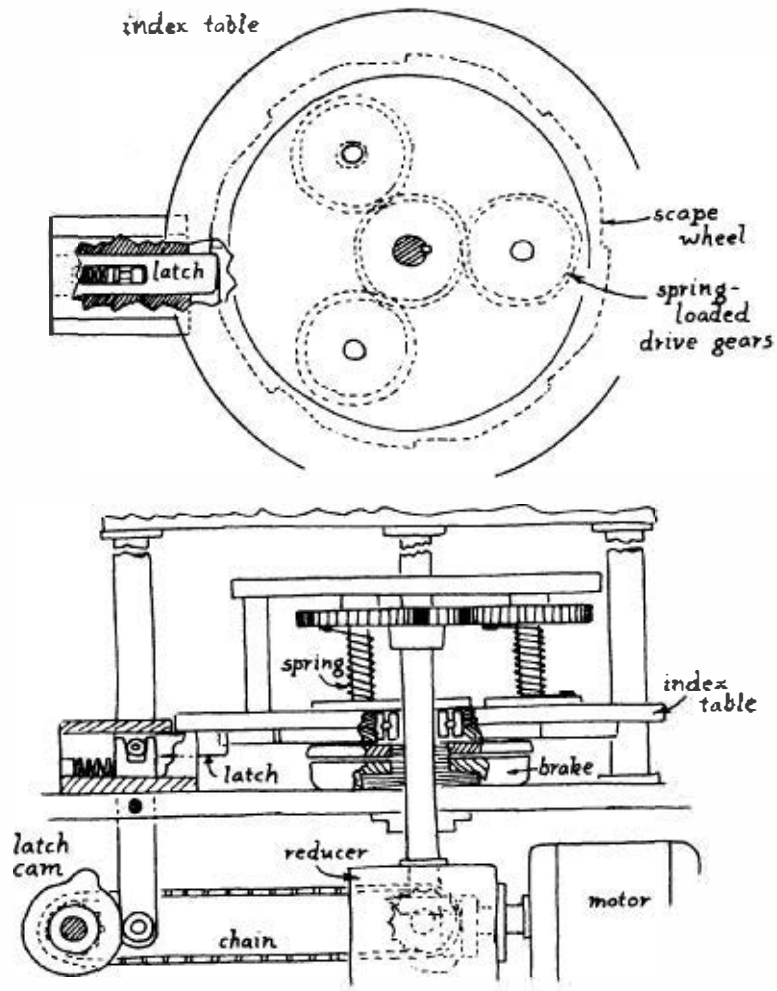


Fig. 12–14. Low-cost load-and-fire escapement.



Developed by J. W. Rieck and A. L. Van Nest; Western Electric Company

Fig. 12–15. Heavy-duty load-and-fire machine escapement for indexing a large machine tool table.

a very simple arrangement in which both dwell and cycle times are controlled by the input.

The heavy-duty load-and-fire machine escapement in Fig. 12–15 is used to index the table of a large machine tool. It is the same type of device as in Fig. 12–14, but with significantly different details. The motor (lower right) winds three large coil springs which are connected to the indexing table through gears. The motor also drives a cam (lower left) which eventually operates a spring-loaded latch releasing the table for one step.

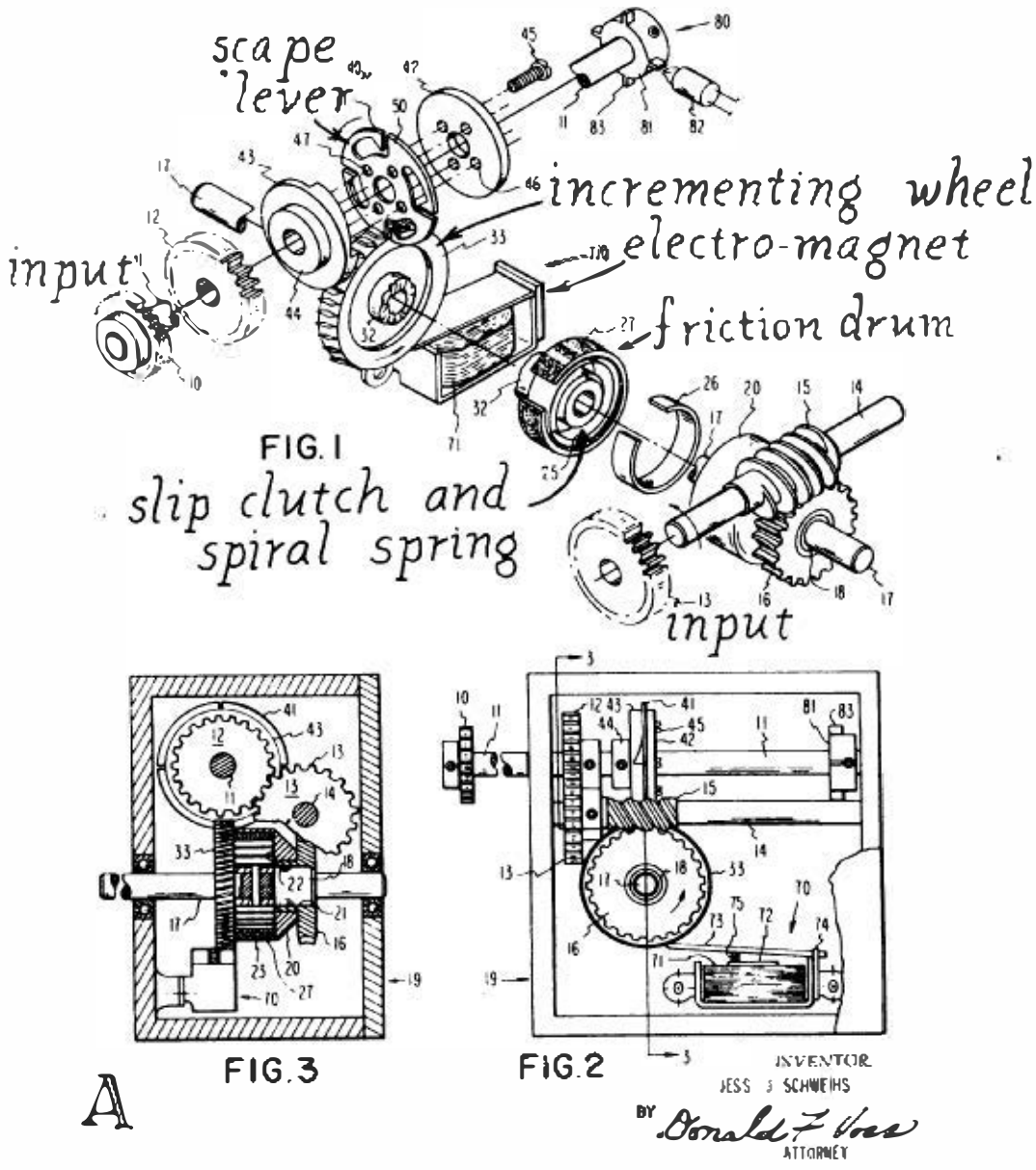
Another load-and-fire mechanism is this escapement (Fig. 12–16A), from a business machine. The "incrementing" wheel (we would have called it a scape wheel) is continuously urged to rotate by a spiral spring (*25*), which is kept wound by a friction drum (*27*). The incrementing wheel, however, is prevented from rotating if solenoid (*70*) is de-energized. A wheel with *flexible teeth* (*50*), serves as the scape "lever." It rotates continuously, periodically escaping the incrementing wheel as long as the solenoid is energized. When the incrementing wheel is held by the solenoid the scape lever wheel (*50*), can still rotate because it has flexible teeth as shown in Fig. 12–16B. Input to this wheel and to the spiral spring, therefore, can be continuous.

A drive roller in the three-piece escapement shown in Fig. 12–17, is frictionally connected to a small cam which contains a wedging tooth. A pair of spring arms constitutes the scape lever (designated as "escapement" in the illustration). When the scape lever is moved toward the left, the drive roller rotates the cam counterclockwise until a resilient wedging tooth contacts the output cam. The wedging tooth and then the drive roller will now drive the output cam until the cutout in the output cam is again positioned opposite the drive roller, as shown in the illustration. At this point motion ceases. The wedging tooth by this time has moved past engagement with the output cam and the tooth carrier has been trapped by the right-hand arm of the scape lever. Moving the scape lever to the right releases the cam to return to the position it had at the beginning of the cycle. This escapement, then, produces one revolution in the output every time the escapement lever is actuated.

Two electrical coils are used to aid permanent magnets in this escapement from a high-speed business machine, Fig. 12–18. An iron scape bar mounted on the end of the spring is captured by the upper or lower permanent magnet until one of the electrical coils is energized; at which point the scape bar will be released by one magnet and captured by the other. Energizing the coils in the proper sequence, therefore, moves the scape bar rapidly up and down, thus releasing the output drum.

The input member of the controlled-output escapement shown in Fig. 12–19 not only drives the scape wheel but releases it as well, and thereby controls output speed, which is rare in an escapement. The scape lever is driven out of engagement with the scape wheel by the rotation of the scape wheel, and is again locked into engagement with the scape wheel during dwell periods by the input cam. This is apparently quite an old design (see Fig. 7–2 for an example).

Figure 12–20 shows a breakaway drawing of a business machine escapement. This highly developed mechanism was designed for the IBM Selectric typesetter. As with the ratchet mechanism of Fig. 7-41,



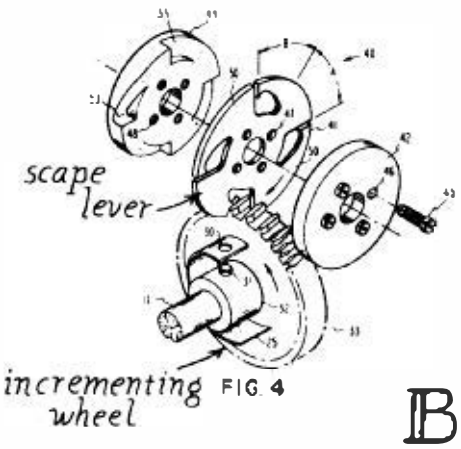


Fig. 12-16. Business machine escapement. Note how in detailed diagram B, at right, portions of scape lever flex to engage teeth in incrementing wheel. (U.S. Patent 3,443,442; J. J. Schweihs.)

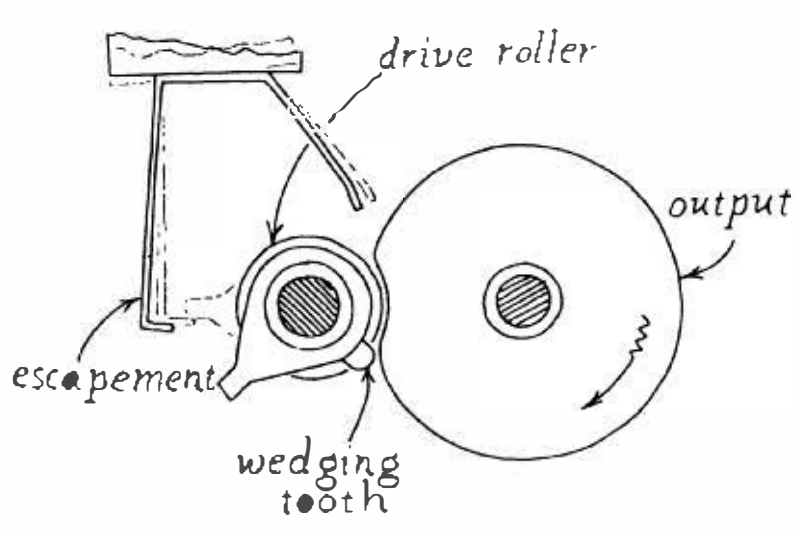


Fig. 12-17. Three-piece escapement with friction drive. (U.S. Patent 3,446,086; J. R. Peltz, et al.)

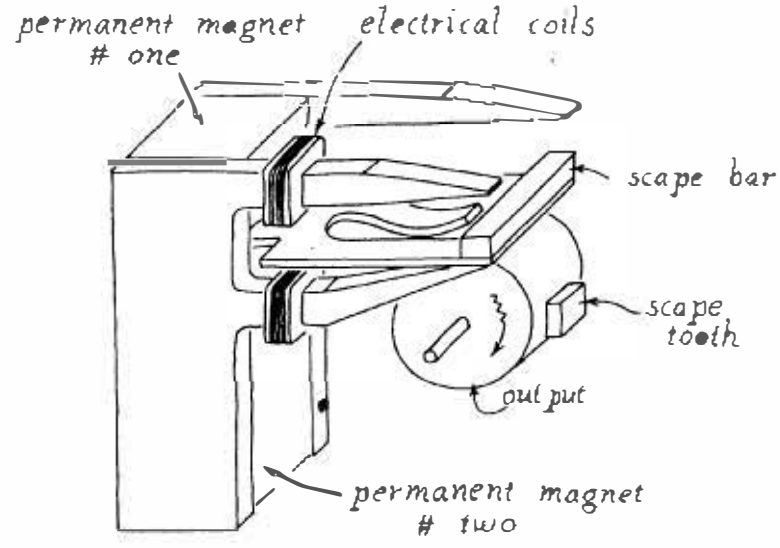
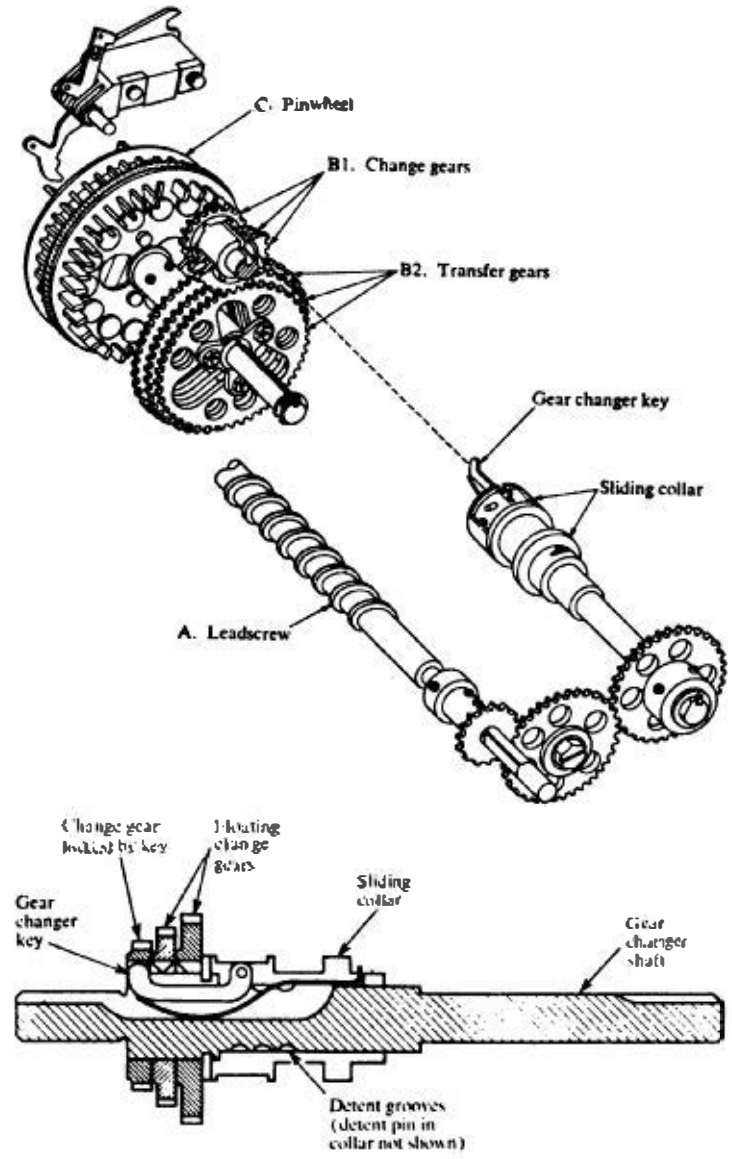


Fig. 12-18. Escapement from a high-speed business machine. (U.S. Patent 3,432,695; J. G. Sims, Jr.)

it was necessary to provide for motion steps of varying lengths. This has been accomplished, in part, by providing change gears to control the ratio between the motion of the carriage lead screw and motion of the escapement. The change gear arrangement, with the sliding collar "clutch" used to couple one of three gears to the gear changer shaft, is best shown in Fig. 12–20. The carriage lead screw can also be seen in this illustration. To provide additional variation in the length of the motion steps, the teeth of the scape wheel were made of sliding pins rather than fixed pins, which can best be seen in Fig. 12–21A. Each pin can be moved laterally out of engagement (or into engagement) with the single-pallet scape lever. The mechanism which controls the pins is shown in Fig. 12-21B; the scape lever is shown in Figs. 12-21A and 12-21C.

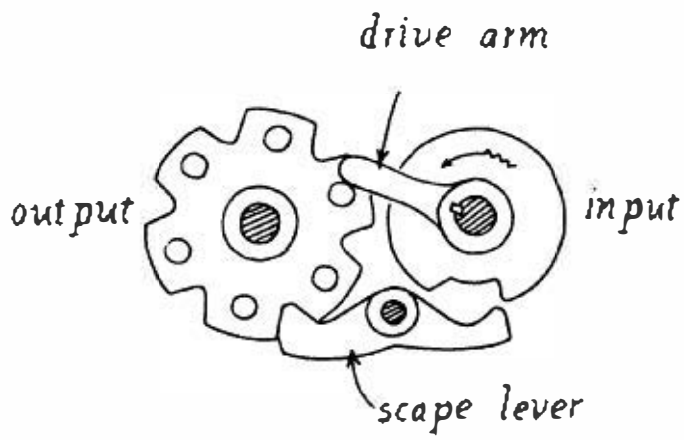
The pin set and clear bars shown in Fig. 12-21B are controlled by logic circuits in the machine, so that different motion steps can be taken for different type fonts and for different characters: a "w" would require a longer step than an "i", for example.

The escape lever in Fig. 12–21C shows the type of refinement that goes into the design of high-speed, high-quality intermittent motion mechanisms. Although only one pallet assembly is provided, there are really two pallets; nearly identical and operating beside each other. One of these is free to jump ahead of the other to catch the next extended pin on the scape wheel, cushioning the impact required to stop the wheel, and improving the reliability of the system by giving the principal pallet more time to re-engage the wheel. See also Fig. 12–3 for a similar, but simpler arrangement.



Drawings courtesy of B. W. Miles and C. C. Wilson, IBM Journal of Research and Development; Vol. 12, No. 1, Jan. 1968: pp. 48–59

Fig. 12–20. Business machine escapement.



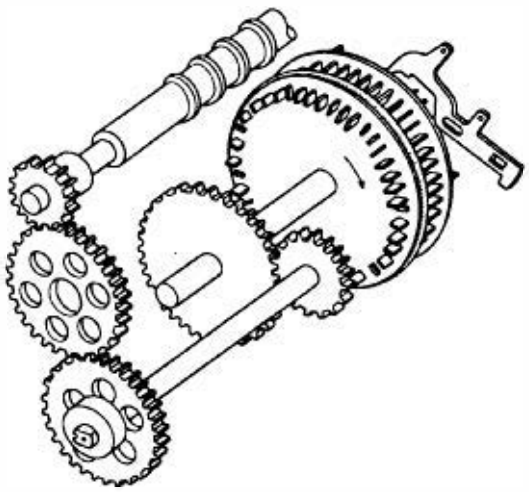
Drawing courtesy of PRODUCT ENGINEERING Magazine; Dec. 20, 1965: pp. 88-93

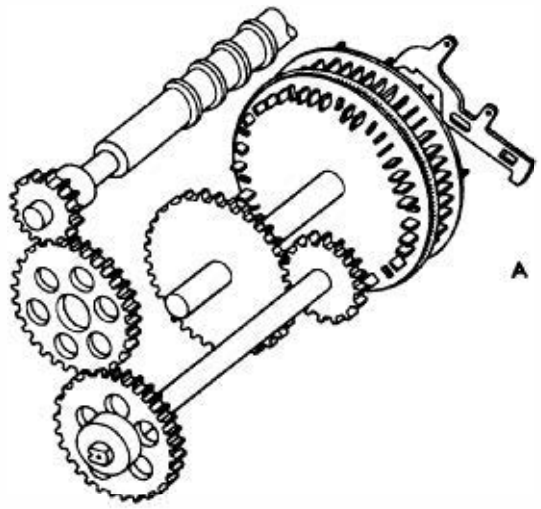
Fig. 12–19. Controlled output escapement.

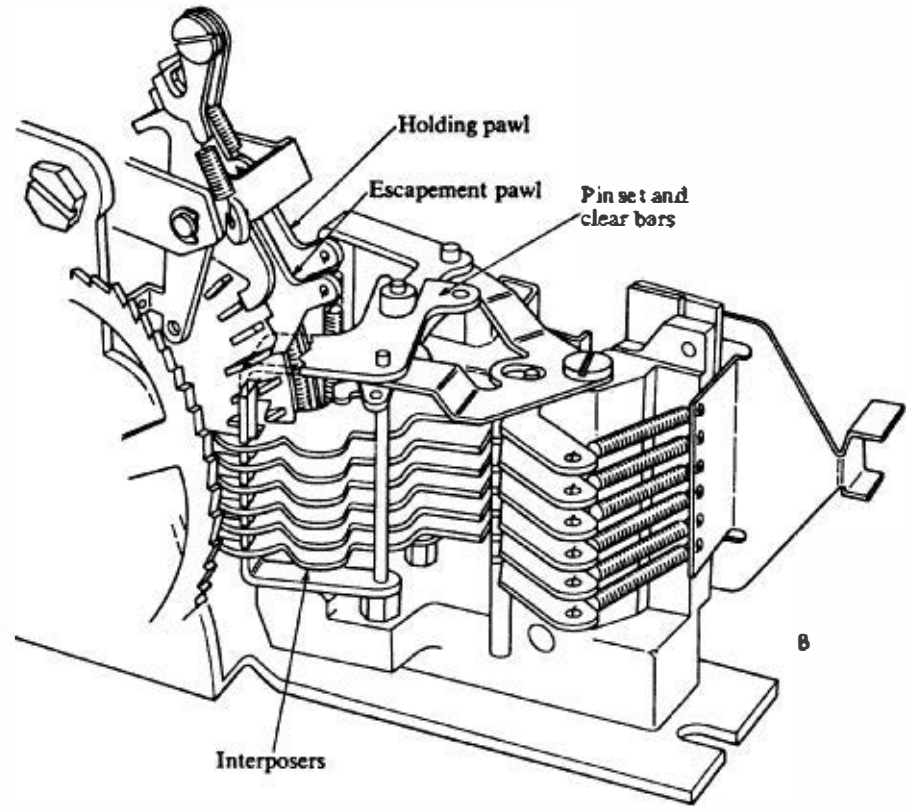
INVERSE ESCAPEMENTS

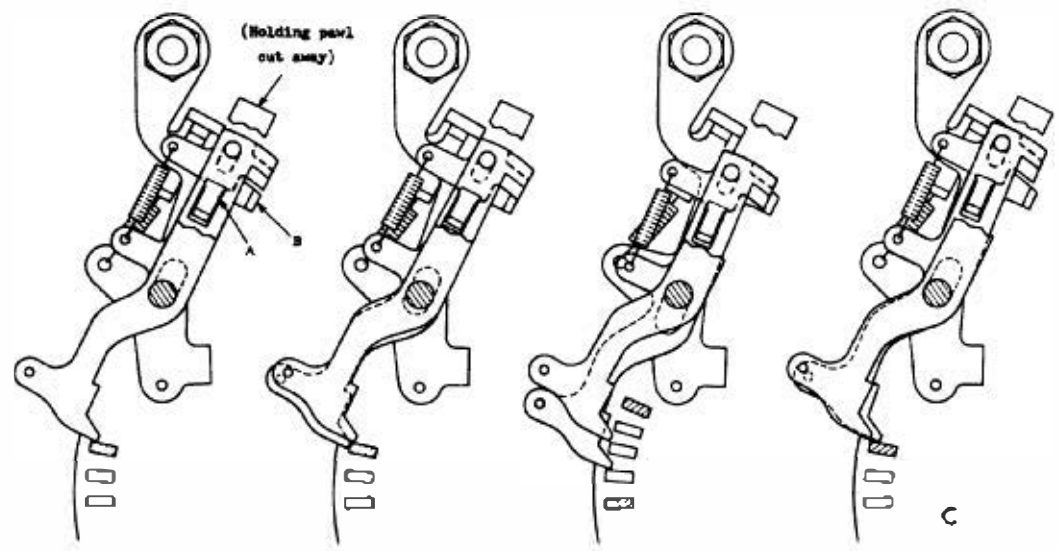
Description of Inverse Escapement

Figure 12–22 shows a typical inverse escapement designed for an instrument application; in this case, a high-speed electrical counter. At first glance this looks like a machine escapement with the scape lever being controlled by a solenoid. Notice the size of the solenoid, however. In the inverse escapement the power flow has been altered: the verge arm (or scape lever) drives, rather than releases, the ratchet wheel. All input energy is derived from the drive solenoid and return spring. I have called the "scape wheel" a ratchet wheel, to further emphasize that this system operates more like a ratchet than an escapement.









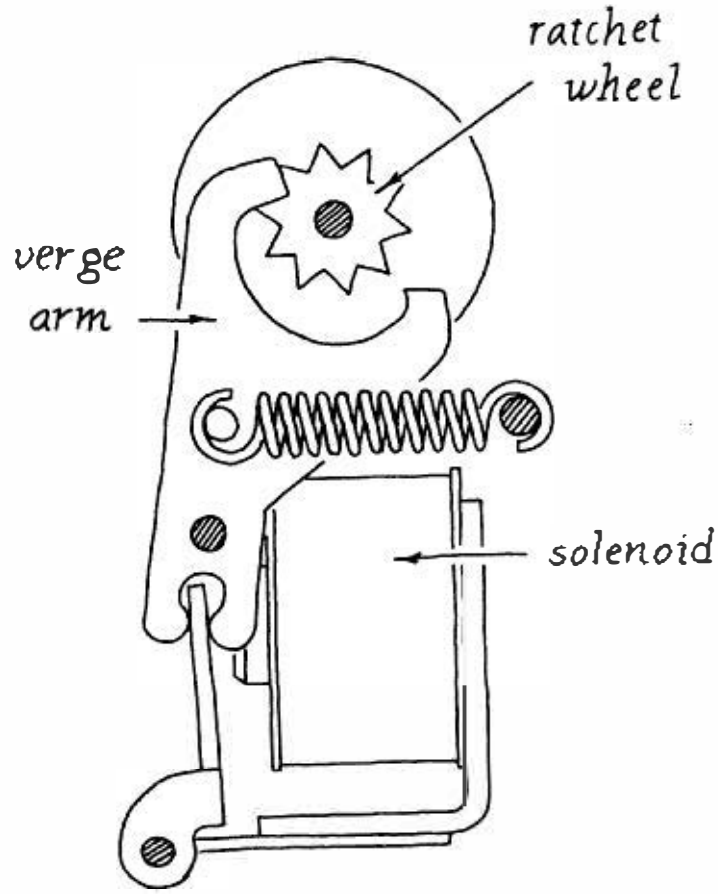
Drawings courtesy of B. W. Miles and C. C. Wilson. IBM Journal of Research and Development; Vol. 12, No. 1, Jan. 1968: pp. 48–59

Fig. 12–21. Details of business machine escapement in Fig. 12–20.

The inverse escapement functions very much like a ratchet but it is simpler than a ratchet. Notice the absence of no-back and non-overthrow pawls, for example, which are required in a ratchet to improve control. In a properly designed inverse escapement, just as in a machine or a clock escapement, one pallet of the verge or scape lever is always in position to interfere with the free rotation of the output wheel which, as a result, can never get ahead of or behind the intended drive position. The inverse escapement, of course, is a cam system in which the "follower" is driving the cam. Like the machine escapement, it is not a very popular device, but it should be more popular than it is. I am sure there are many machine designs now using ratchets that could be improved by the use of an inverse escapement.

Advantages and Disadvantages of Inverse Escapement

The inverse escapement is a very simple device. As a result, its reliability and cost are probably more attractive than those of a ratchet, which is its nearest competitor. It is probably a little better at load control than a good ratchet (by "good" ratchet is



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 12–22. Typical inverse escapement.

meant one with proper non-overthrow teeth, no-back pawl, etc.); especially at very high speeds. It is possible, however, for the ratchet wheel on the inverse escapement to rebound and back up if it is forced to operate at too high a speed.

Like other escapements, of course, this one produces impact, probably to about the same extent as a ratchet. This means that it is usually used only in relatively light-duty applications, although unlike a machine escapement, the inverse escapement can handle heavy loads at slow speeds as we shall see later in Figs. 12–33 and 12–35. The steps it takes must be fairly short, so the motion times are relatively short and cannot be extended. The necessity to maintain a good cam angle between verge and output wheel (to guarantee adequate drive torque) also limits the number of stops which this device can make per output revolution. (See Fig. 8–3 for further discussion of this point.)

It is doubtful that a successful inverse escapement could be built which would take fewer than two, or three, or more than 100 steps per revolution. However, it is always possible that a clever designer could find a way around any flat statement of that sort! But, at least, such has been my experience.

Motion Curves of Inverse Escapements

The motion curves of inverse escapements are essentially the same as those of impulse ratchets as will be seen later, in Fig. 12–23. When the verge arm strikes the ratchet wheel, the wheel will start to rotate but the arm will rebound. The solenoid brings the arm back under control and moves it back into the wheel for a second blow. After four or five separate impacts of this kind, the arm and wheel will move together to complete the stroke. When the arm has reached the bottom of the wheel tooth, the wheel will come to rest. Since there is some play between wheel and drive teeth, and since the arm will rebound again at the end of the stroke (increasing the backlash a little), the stopping process will also consist of a series of impacts as shown in the curves.

Figure 12-23 depicts motion curves for a typical inverse escapement. Note that these are the same

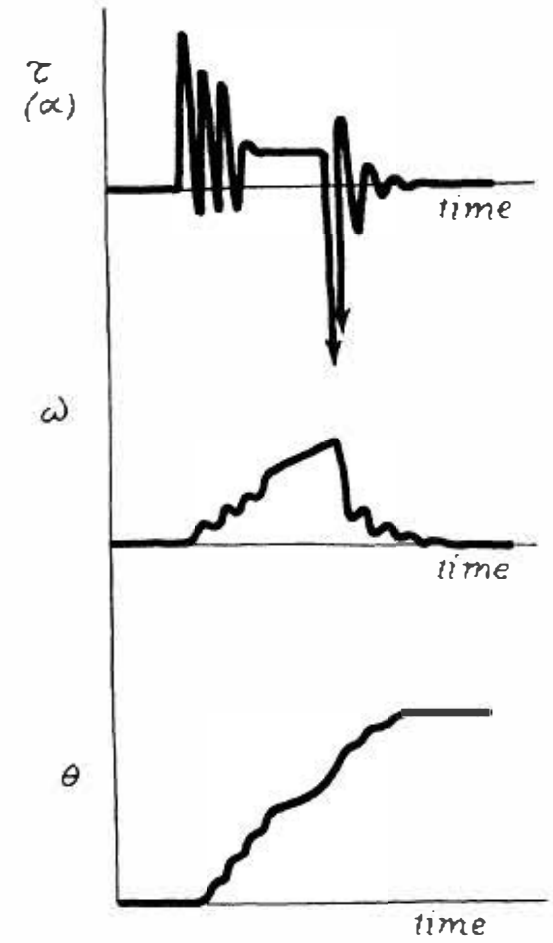
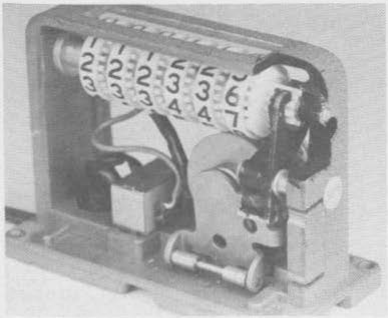


Fig. 12–23. Motion curves for a typical inverse escapement.



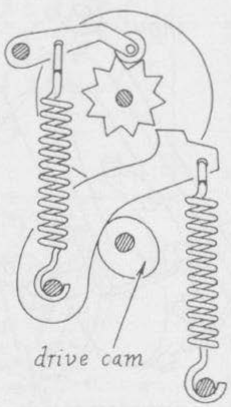
Photograph courtesy of the Veeder-Root Company

Fig. 12–24. Electrical counter driven by the inverse escapement of Fig. 12–22.

as the curves for an impulse ratchet as shown in Fig. 7–10.

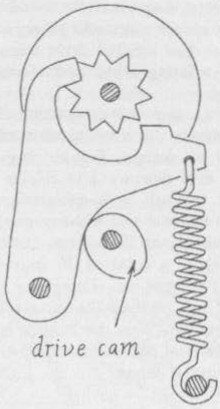
In Fig. 12–24 the photograph shows an electrical counter driven by the inverse escapement of Fig. 12–22. A portion of the frame of the counter has been removed to reveal the driver more clearly.

In the design of a single-pallet inverse escapement, seen in Fig. 12–25, a continuously rotating input cam operates a single-pallet verge. Note that a detent roller is required to control the output during dwell periods since only a single-pallet is used.



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965: p. 121 ff

Fig. 12–25. A single-pallet inverse escapement.

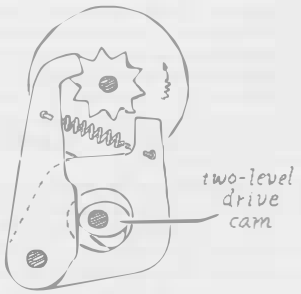


Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965: p. 121 ff

Fig. 12–26. Inverse escapement with continuously rotating input.

Otherwise the same as Fig. 12–22, the inverse escapement shown in Fig. 12–26 has a continuously rotating input.

There is a period during each drive stroke in an inverse escapement when the wheel is very loosely controlled, although it is not completely free. This can cause troubles which can be avoided, however, if each pallet is separately driven so that the "holding" tooth could be left in engagement with the output wheel until the next "drive" tooth has com-



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965: p. 121 ff

Fig. 12–27. Inverse escapement with continuously rotating input and separate drive for each pallet

pleted its pre-travel and is ready to index the wheel. The two-level cam arrangement shown in Fig. 12–27 would produce this kind of drive sequence; i.e., a continuously rotating input and a separate drive for each pallet.

In a bidirectional inverse escapement (Fig. 12–28), when the clockwise coil is energized it will rotate the output wheel 18 degrees, first driving the spring-loaded lower arm downward to release the output wheel. When the coil is de-energized, the spring loaded arm is released and will engage the wheel to drive it an additional 18 degrees, thereby turning the output wheel a total of 36 degrees for each excitation of the input coil. Energizing the counter clockwise coil will produce the same results but in the other direction. This mechanism is used in a high-speed electrical counter made by the Citizens' Watch Company of Japan.

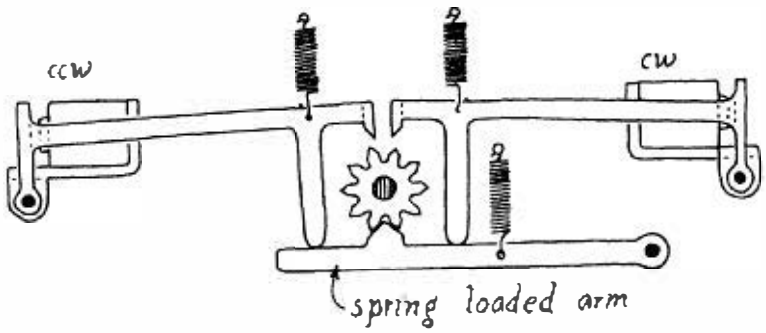


Fig. 12–28. Bidirectional inverse escapement.

A low-cost inverse escapement with magnetic drive for the second half of each stroke is shown in Fig. 12–29. When the input cam pulls down the spring the magnet (which is located behind the wheel) will rotate the iron output wheel a few degrees counterclockwise, since the magnet is nearer to tooth *a* than to tooth *b*. When the cam releases the drive spring it will move upward and cam the wheel into the next position, corresponding to that shown in the illustration, to complete the drive stroke. (U.S. Patent 3,504,306; D. Fritsch.)

The motion of the side-acting inverse escapement device in Fig. 12–30 is basically the same as that of Fig. 12–22, but in this case the action is sideways. To see why these devices are called "escapements" even though they are drivers, compare this drawing with Fig. 11–7.

Figure 12–31 illustrates an axial inverse escapement. The three, toothed members in this illustration are cylinders. The center one is prevented from ro-

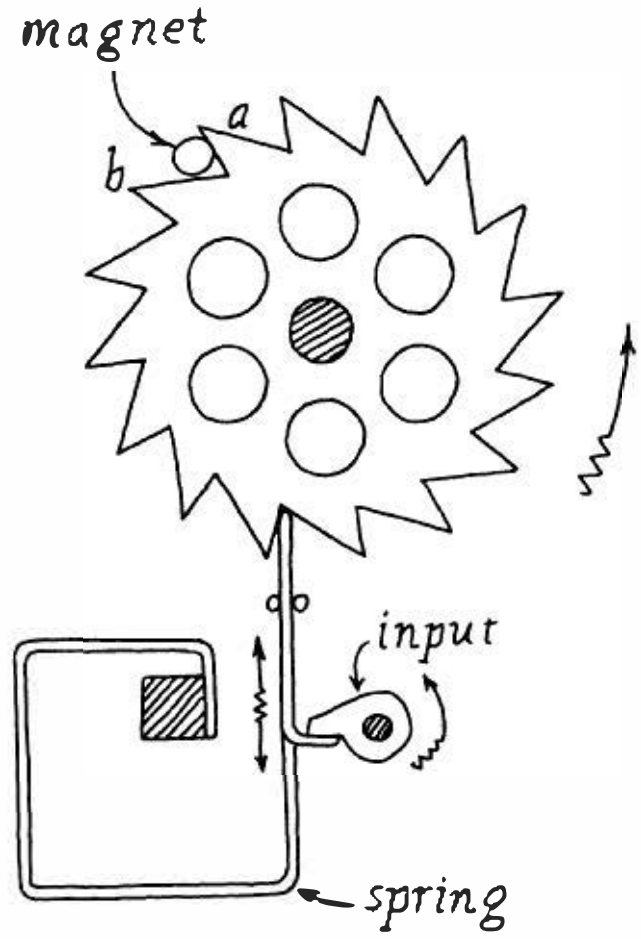
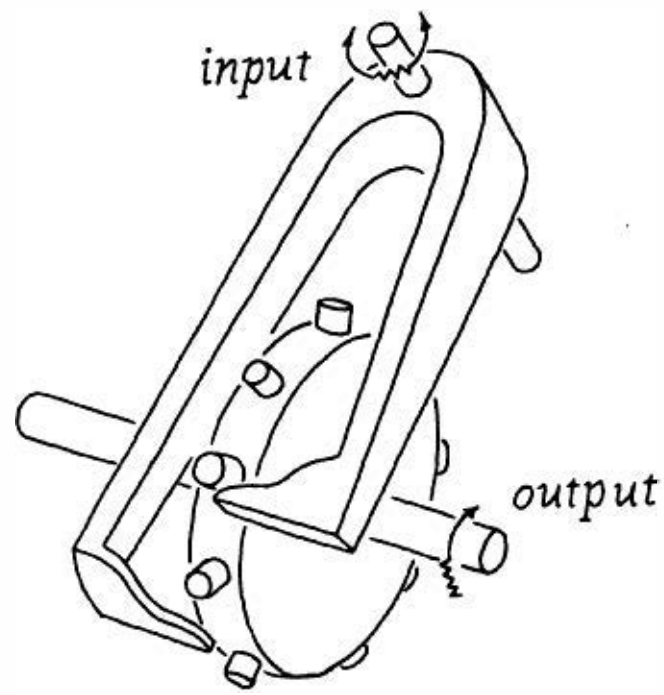


Fig. 12–29. Low-cost inverse escapement with magnetic drive for the second half of each stroke. (U.S. Patent 3,504, 306; D. Fritsch.)



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 22, 1966; p. 121 ff

Fig. 12–30. Side-acting inverse escapement.

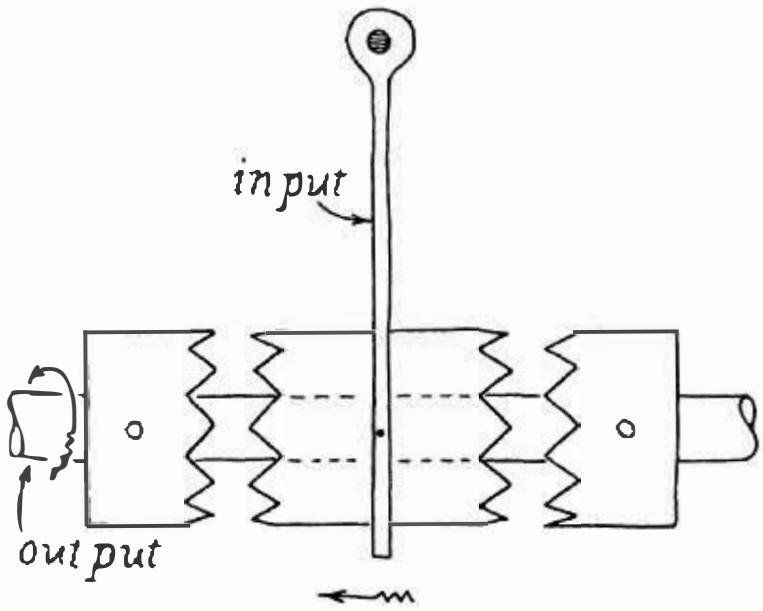


Fig. 12–31. Axial inverse escapement.

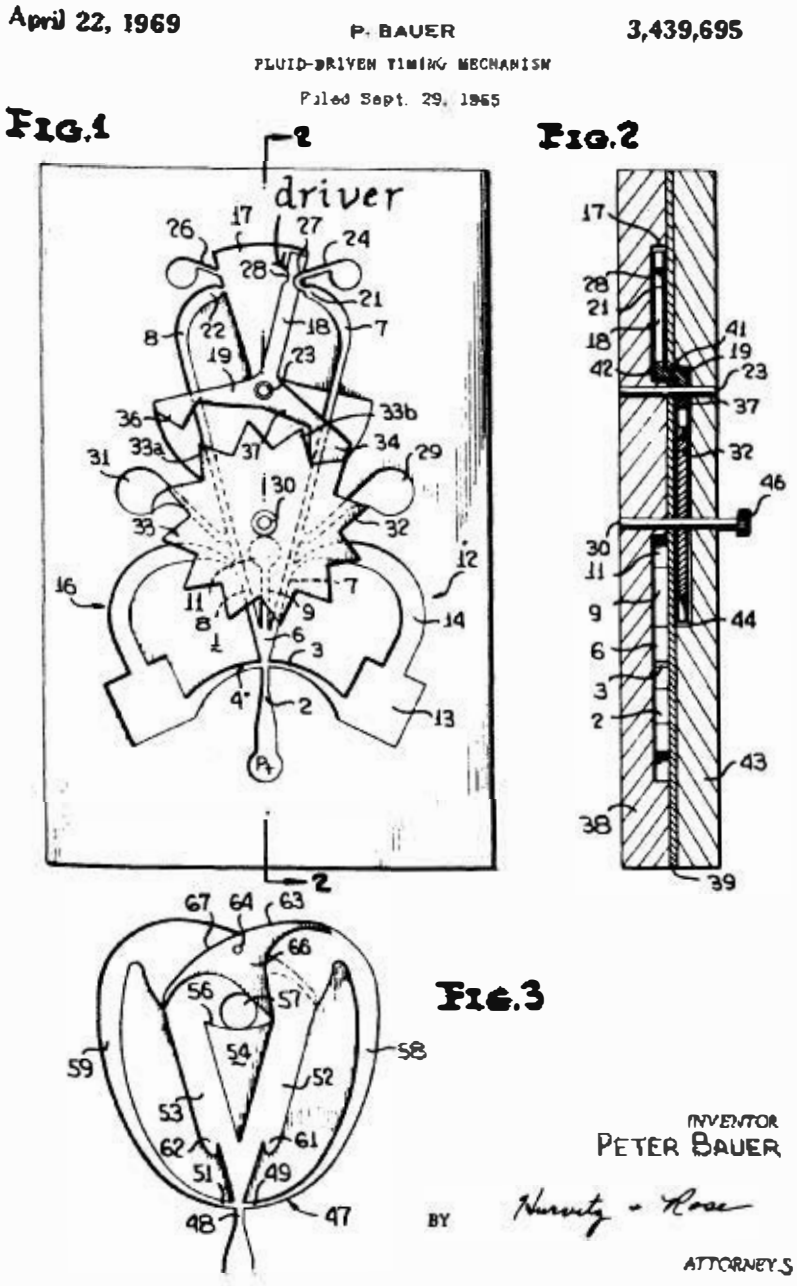


Fig. 12–32. Fluidic inverse escapement. (U.S. Patent 3,439,695; P. Bauer.)

tating, and is moved back and forth, as shown, alternately engaging the cylinders on the left- and right-hand ends of the shaft, both of which are pinned to the shaft. This indexes the shaft.

The principle of the fluidic inverse escapement shown in Fig. 12–32 is the same as that of Fig. 12–22, but here the driver (*28*) is powered by jets of fluid rather than by a solenoid.

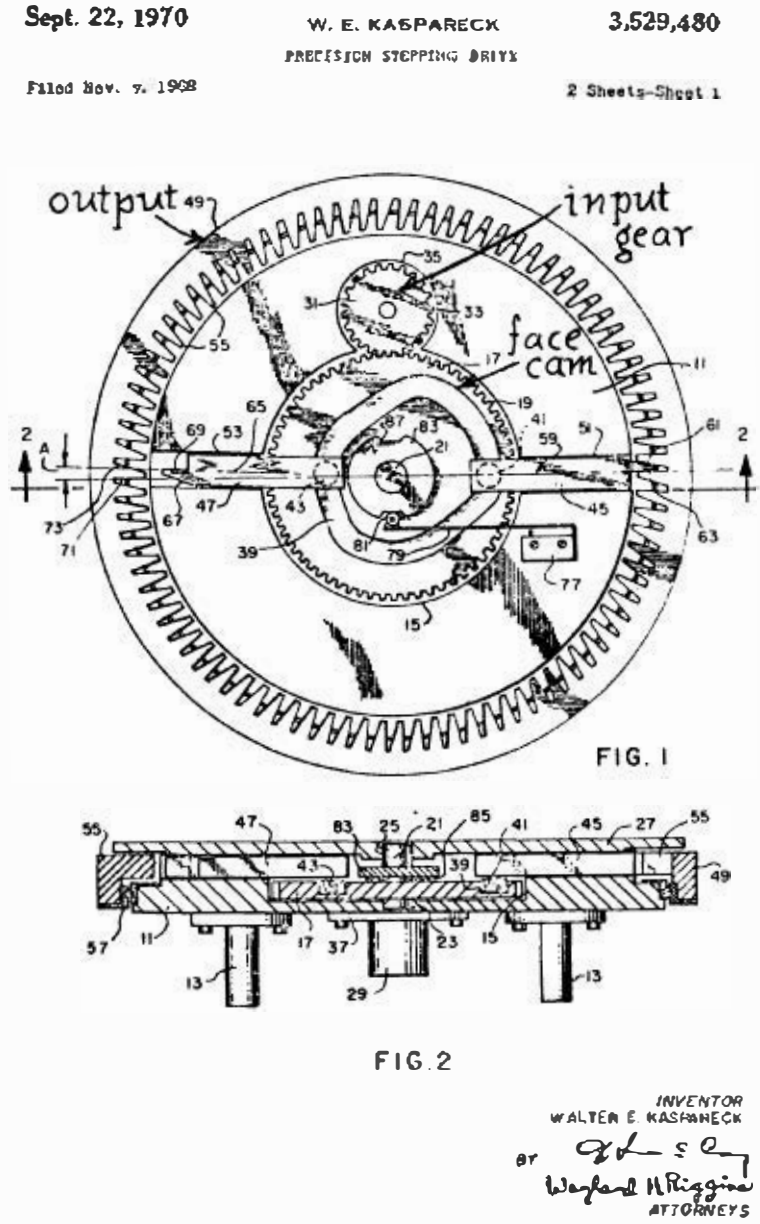


Fig. 12–33. Inverse escapement for an aerospace application. (U.S. Patent 3,529,480; W. E. Kaspareck.)

The design for the inverse escapement for an aerospace application shown in Fig. 12–33 was conceived at the National Aeronautics and Space Agency. Input is through gear (*31*) which rotates face cam (*17*) to move the drive arms in and out of engagement with the teeth of the output member (*49*). NASA says this device produces better torque with greater precision than a conventional gear train. The sequence of operation of the inverse escapement

is shown in Fig. 12-34. A design such as this could handle a very heavy load but only at a relatively slow speed.

In the inverse escapement of Fig. 12-35, used on a heavy-duty machine tool, the drive member (*21*) reciprocates rather than rotates, but the principle is still basically the same. Again, this escapement would

Sept. 22, 1970 W. E. KASPARECK 3,529,480

PRECISION STEPPING DRIVE

Filed Nov. 7, 1968 2 Sheets-Sheet 2

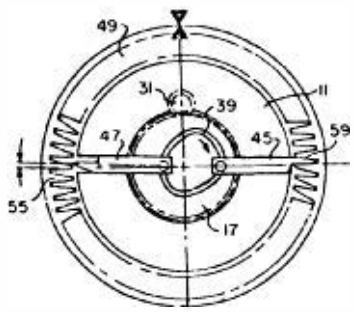


FIG. 3

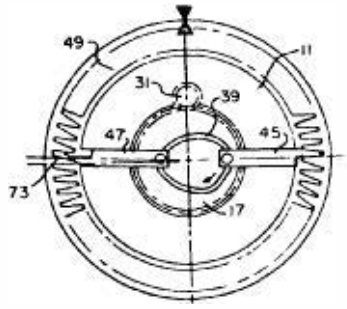


FIG. 4

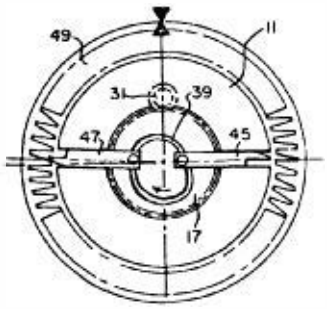


FIG. 5

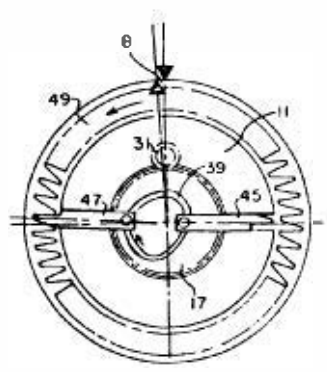


FIG. 6

INVENTOR
WALTER E KASPAREK
BY
Wayland H Riggin
ATTORNEYS

Fig. 12-34. Sequence of operation of stepping drive shown in Fig. 12-33. (U.S. Patent 3,529,480; W. E. Kaspareck.)

be suitable for heavy loads if the speed were low enough.

Among the miscellaneous inverse escapements from high-speed electrical counters in Fig. 12-36, notice the similarity between sketch *D* and Fig. 12-35.

The doublearm inverse escapement in Fig. 12-37 looks less like an escapement and more like a cam

Dec. 15, 1964 H. J. VENABLES III 3,161,070

INDEXING APPARATUS

Filed Jan. 28, 1964 2 Sheets-Sheet 1

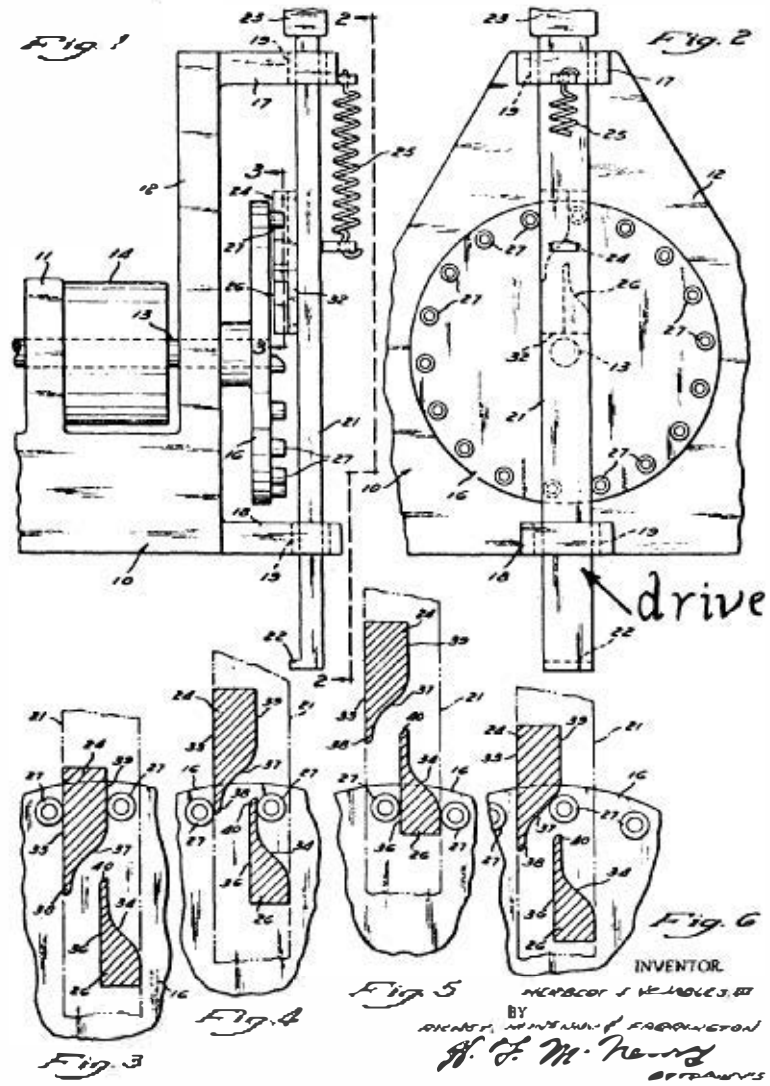
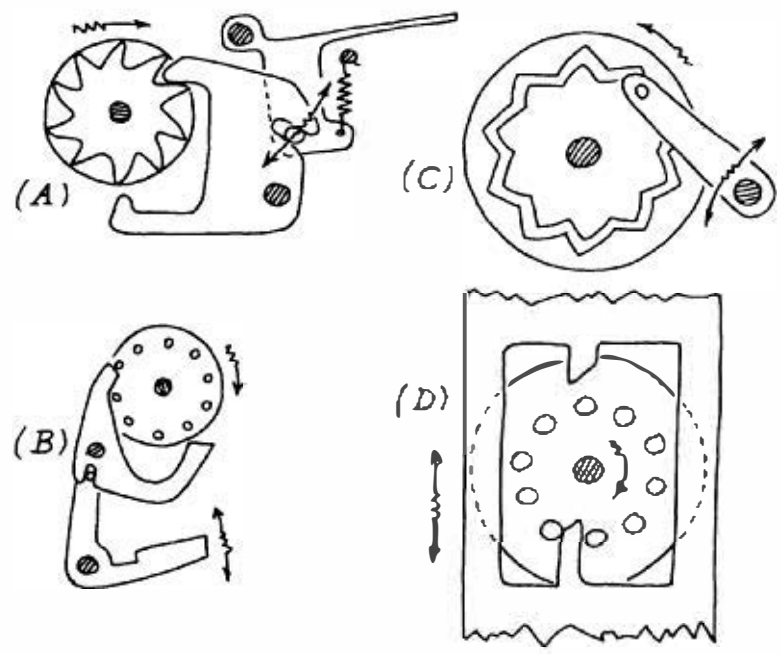


Fig. 12-35. Reciprocating inverse escapement used on heavy-duty machine tool. (U.S. Patent 3,161,070; H. J. Venables, III.)



Drawings courtesy of the Veeder-Root Company

Fig. 12-36. Miscellaneous inverse escapements used in high-speed electrical counters.

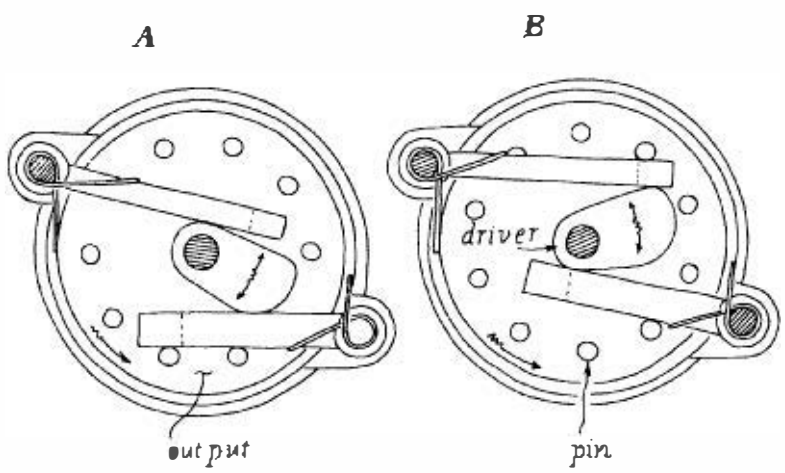


Fig. 12-37. Double-arm inverse escapement. (U.S. Patent 3,444,764; R. Greenwood.)

system than most of the devices in this chapter, but it combines the features of Figs. 12-36D and 12-27. As in 12-36D, for example, the output cam contains a circle of drive pins. These are pushed alternately by two separate drive arms, as was the design of Fig. 12-27. In Fig. 12-37, however, the input cam oscillates, rather than rotates. This motion forces the drive arms outward, alternately, engaging the pins on the output wheel. Note that the drive arms have drive "blocks" at their ends; the pins on the output wheel are able to move under the inner portion of the arms as the wheel turns. The mouse trap springs force the drive arms to follow the input cam.

Anatomy of an Inverse Escapement

Let us now review the component parts of an inverse escapement device as seen in Fig. 12-38:

A. Output wheel—joined to *C*; B. Verge or drive arm—serves as drive pawl, non-overthrow pawl, and no-back pawl making this mechanism simpler than a ratchet having the same speed capability; C. "Ratchet" or "star" wheel—teeth must be shaped to provide proper cam angles for the forces received from the verge. It is the principal wear point; D. Drive solenoid—attracts the clapper, which moves the verge, which indexes the wheel. It would take another book to describe all the factors which could be considered in the proper design of a relay movement such as this. Such factors as the coil constant, the time constant, eddy current losses, I^2R losses, form factor, residual magnetic effects, bounce, flux levels, leakage losses, temperature rise, air gap, etc., all influence performance; E. Return spring—with a ratchet we can choose direct drive or spring drive. With an inverse escapement, alternate steps are spring driven and power driven. The spring (which is loaded by the solenoid during the power step) must be strong enough to move the verge, wheel, and clapper fast enough to maintain the desired stepping rate; but the spring must be kept as light as possible to minimize power requirements, impact stress, and wear rates; F. Air gap—usually must be adjusted in production if optimum performance is desired. It affects such things as power consumption, operating speed, impact levels, and ultimately (after wear), whether or not the solenoid driver produces enough motion in the output wheel to put the wheel in the correct position to receive the next blow from the next verge tooth; G. Clapper hinge—must not be in the magnetic circuit since magnetic flux has a way of "gluing" moving parts together; H. Bearings—must be large

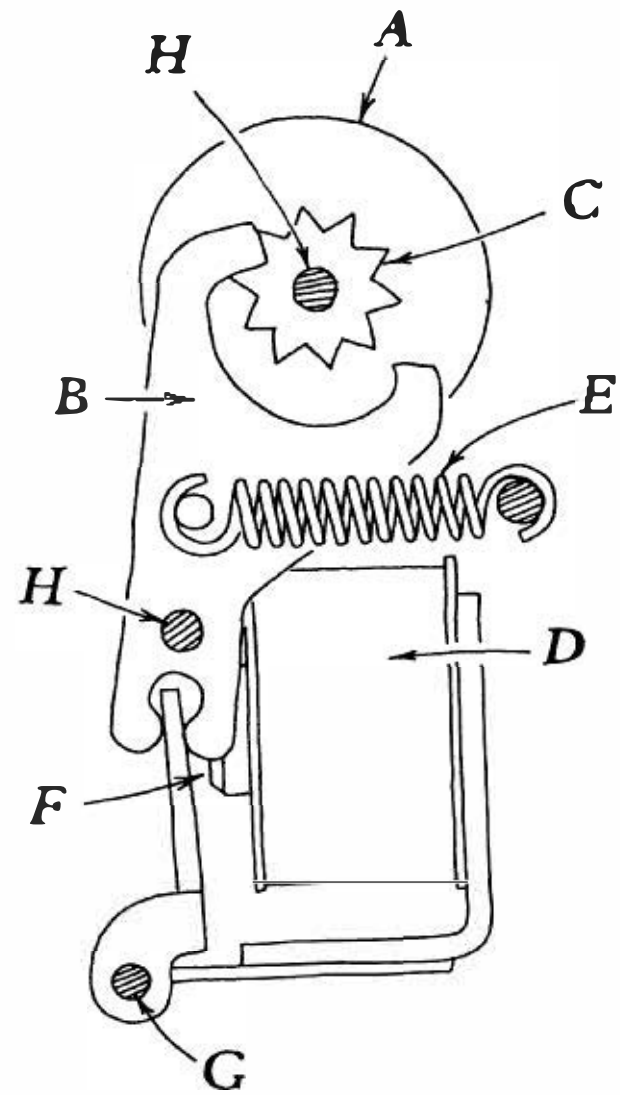
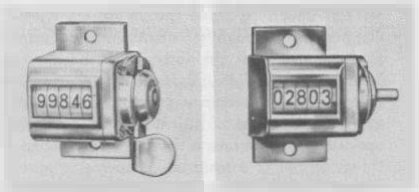


Fig. 12-38. Anatomy of an inverse escapement.

enough in diameter to withstand the repeated impacts to which this system is exposed; yet small enough in diameter to minimize bearing friction torque.



Photograph courtesy of the Veeder-Root Company

Fig. 12-39. (*Left*) Small mechanical counter designed and built by Curtis Veeder in [illegible] of the same counter

Inverse escapements are widely used today in electro-mechanical counters, having almost replaced ratchets for most applications. Stepping motors are also used in some instances, however, when extreme speed and very long life are required. Mechanical counters generally rely on intermittent gearing of some sort. Figure 12-39 shows a small mechanical counter designed and built by Curtis Veeder in 1895; and also a modern version of the same counter. This latter uses the same mechanisms as its predecessor, but is constructed with the newest materials and techniques. The contemporary version was used in [illegible] flights to the moon. From 1895 to [illegible] pressive "life" for a single design.

CHAPTER 13

Clutch and Brake Systems

Systems Versus Components

Most of the mechanisms that have been discussed up to this point produce intermittent motion without any assistance from other components. A Geneva, for example, converts the continuously rotating motion of an input shaft into intermittent motion to drive a load. No external controls, or the like, are needed for the Geneva to perform its designed task. On the other hand, clutches and brakes, are components which, taken alone, usually cannot produce intermittent motion. They are used instead as building blocks of larger systems involving control components, etc.; without these additional components the clutch-brake combination will produce either continuous motion or no motion.

A clutch is a device used to connect one shaft to another, usually connecting a load to a driver on command. A brake is a device used to connect a shaft to ground for stopping or holding that shaft. Usually it is the load shaft which is provided with the brake. Clutches and brakes are very similar devices and, in many cases, are identical to each other in construction. They can be obtained as separate components and provided with separate and independent controls by the system designer. Since they are so frequently used together, however, they can also be purchased as combined clutch-brake components and often with predesigned control packages. Examples of these various products will be seen in this chapter.

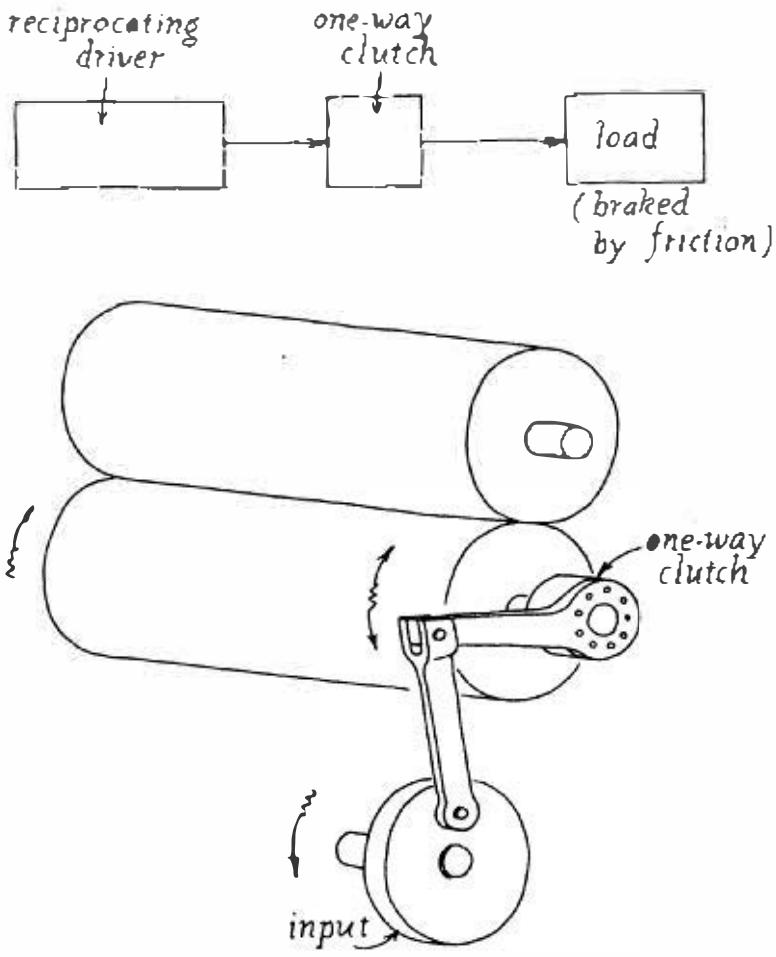
Types

There are many different types of clutches and brakes. It would be possible, in fact, to write a separate book on this subject alone. I hope it will be sufficient for our purposes to say that some clutches and brakes have toothed mating surfaces; some rely on friction (disc or spring clutches and brakes); some make use of wedging or toggle forces (and friction) to grip the output member (sprag, roller, or cam clutches and brakes); and some rely on magnetic or electrical forces to produce the drive or braking torque (hysteresis, eddy current, magnetic powder and magnetic fluid clutches). Most of these various types can be actuated by either mechanical, electrical, pneumatic, or hydraulic means, depending on the system in which they are used.

Systems

Clutch-brake combinations are used in two basically different ways in intermittent motion drives. They are used alone (with suitable control means) to produce intermittent motion where high-precision positioning is not required. When high precision is required, they are used with precision mechanical indexers; Genevas, ratchets, star wheels, gears, etc. Their main function in this second case is to extend dwell periods while permitting fast and precise indexing during drive periods.

Figures 13–1 through 13–8 will describe and illustrate next, a number of different basic clutch-brake



Drawing courtesy of The Lovell Corporation

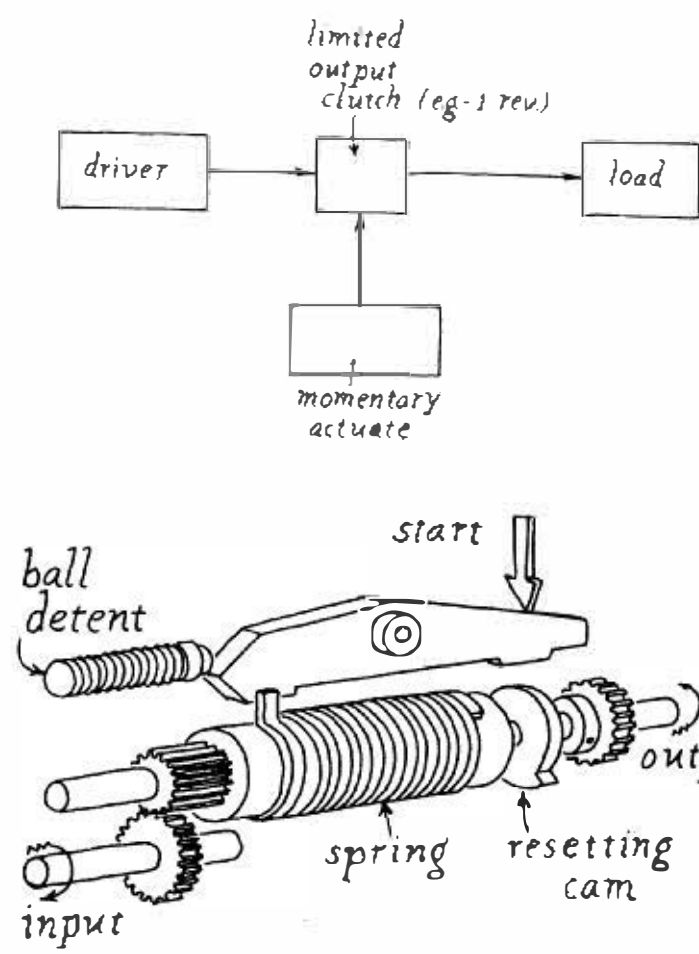
Fig. 13–1. Simple intermittent motion system employing a one-way clutch.

systems for producing intermittent motion. Some of these produce fixed-length steps at a fixed repetition rate. Others produce controllable steps at a controllable rate, while the rest produce fixed steps at a variable rate or vice versa.

Figure 13–1 shows a block diagram and illustration of a simple intermittent motion system employing a one-way clutch. This system produces fixed steps at a fixed rate and no brake is involved since the load is braked by its own friction.

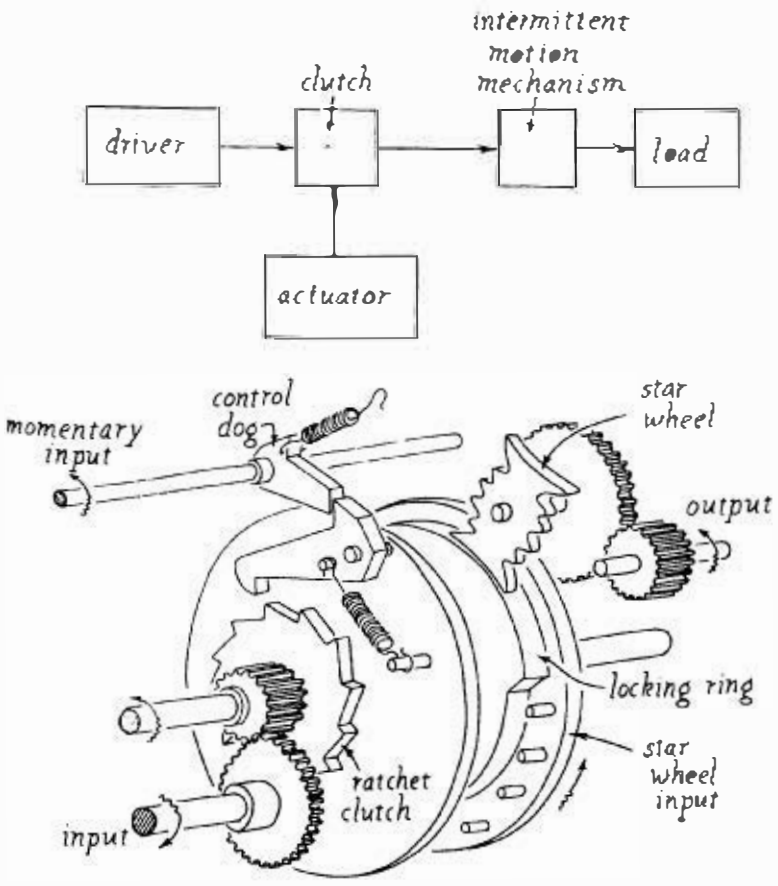
The block diagram and schematic illustration in Fig. 13–2 explain a system in which a momentary control action (pushing on the control lever in the direction shown by the "start" arrow) actuates a clutch. As the control lever is pushed, the left end of the spring is released allowing the spring to grab the input shaft and transmit torque to the output shaft at the other end of the spring. (The barrel-shaft on which the spring is mounted is in two pieces and the spring is fastened to the right-hand end.) As the output shaft turns, it carries with it the resetting cam which brings the starting lever back into its original position (remember, the starting push was only momentary). When the left end of the spring strikes the control lever again, the output motion ceases. And, again, a brake is not necessary in some systems where friction in the load will bring the output to rest in a reasonable time; but it is necessary if load inertia exceeds load friction. This system produces fixed steps with variable dwells.

In the clutch and intermittent motion mechanism system illustrated in Fig. 13–3, a ratchet pawl engages the input ratchet wheel after the momentary rotation of the control input shaft releases the pawl. This produces approximately one turn in the ratchet clutch, which moves until the control dog lifts the ratchet pawl out of engagement with the input ratchet wheel. The ratchet clutch, in turn, drives the star-wheel input, producing precisely 180 degrees of motion in the output shaft each time the system is actuated. This system produces a fixed-length step, but with a variable dwell time (depending on the control system). The brake on the output consists of the locking ring on the star wheel. Other than friction, there is no brake on the ratchet clutch.



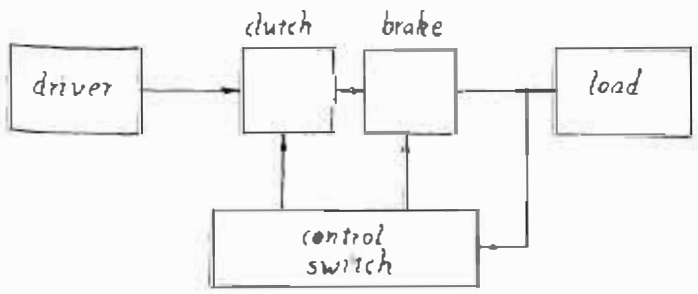
Drawing courtesy of MACHINE DESIGN Magazine, Nov. 21, 1968

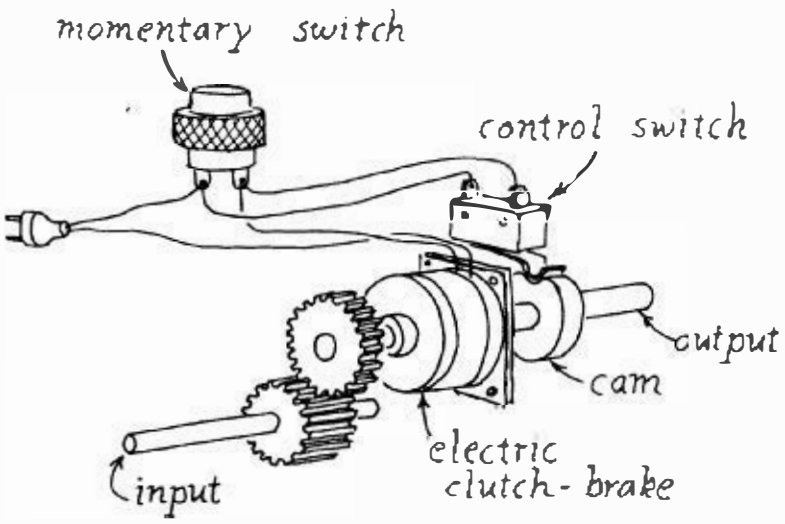
Fig. 13–2. System producing fixed steps with variable dwells.



Drawings courtesy of MACHINE DESIGN Magazine, Nov. 21, 1968

Fig. 13-3. Intermittent motion system with clutch, producing fixed-length step with variable dwell—depending on the control system.





Drawings courtesy of MACHINE DESIGN Magazine, Nov. 21, 1968

Fig. 13-4. Electric clutch-brake system with simple control means.

A block diagram and illustration of an electric clutch-brake system and simple control means are shown in Fig. 13-4. The push-button switch is actuated momentarily, energizing the clutch and de-energizing the brake. The output shaft starts to move. As soon as the output shaft has moved a few degrees, the cam has closed the control switch which overrides the momentary starting switch and keeps the clutch energized and the brake de-energized. After the output shaft has moved one revolution the control switch is opened. If, by this time, the momentary switch has been released by the operator the clutch will de-energize and the brake will stop the output motion. This system produces fixed steps with variable dwell.

The block diagram in Fig. 13-5 shows a self-controlling clutch-brake system. A rotating driver

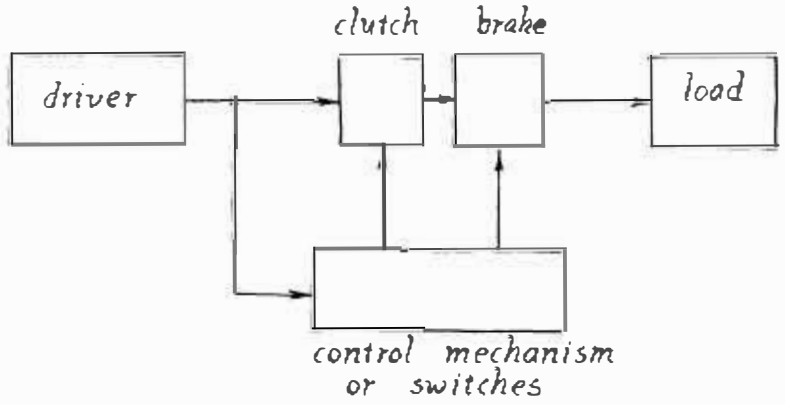
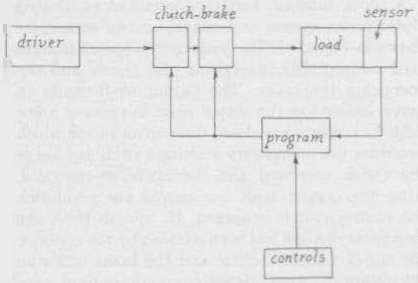


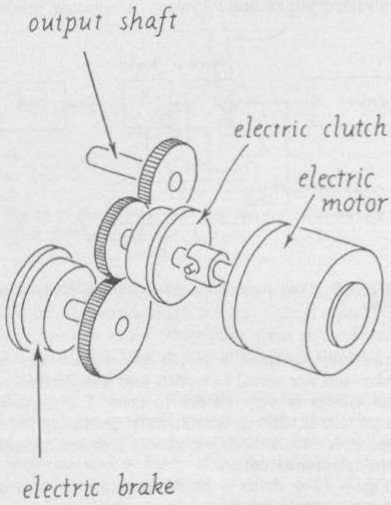
Fig. 13-5. Block diagram of a self-controlling clutch-brake system.

periodically actuates a clutch and de-actuates a brake (and vice versa) to move a load intermittently. This system is very similar to that of Fig. 13-4 except that it will run continuously, producing fixed steps with a fixed dwell period and does not depend upon operator actuation.

Figure 13-6 shows a block diagram (left) of a programmable clutch-brake system. The programmer might be a computer, a bank of switches, a punched card reader, a limit switch and cam system, or the like. The sensor connected to the load could be a shaft encoder, limit switch, pulse generator, etc. The control system might be used to drive a clutch-brake system such as that shown in the illustration at the right. It will produce a fixed or variable step and a fixed or variable dwell, as programmed.

The switch bank in the counter of the programmable clutch-brake system of Fig. 13-7 (in which a predetermining counter is used as the program and



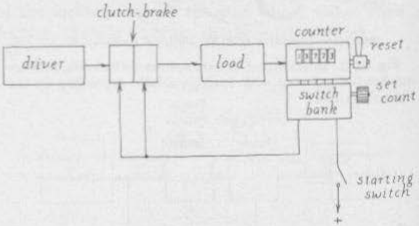


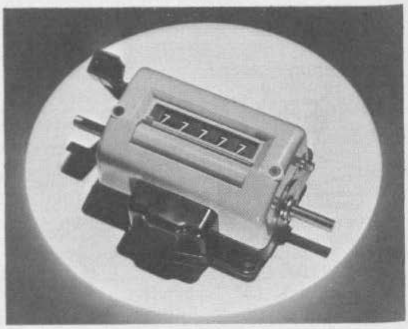
Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 13-6. (*Top*) Block diagram of a programmable clutch-brake system. (*Bottom*) Components of this system.

sensing means), is set at a desired number of turns—175, for example. The counter is then reset to zero. At this point the electrical contacts in the switch bank are all closed. When the starting switch is closed, power is applied to the clutch-brake combination to couple the load to the rotating driver, and motion starts. After 175 revolutions the counter reaches the number set in the switch bank. At this point the contacts in the switch bank open, de-energizing the clutch and energizing the brake to stop the load. The system will remain stopped until the counter is reset. This system will produce a fixed step (although it can be changed) with a variable dwell. The illustration at the bottom shows a mechanically driven predetermining counter.

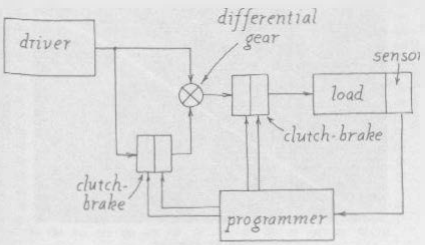
The block diagram in Fig. 13-8 shows a system for starting and stopping a load by progressive acceleration and deceleration. Usually, it is undesirable to start or stop a heavy load suddenly. One way to reduce accelerations and decelerations is to start and stop the load in steps; first bringing it up to half speed (for example), then bringing it to full speed, dropping it to half speed, and finally dropping it to zero. One way to obtain this sequence is to use an adding differential and two clutch-brake systems as

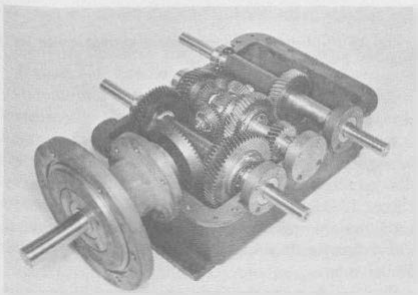




Photograph courtesy of the Veeder-Root Company

Fig. 13-7. (*Top*) Programmable clutch-brake system with predetermining counter used as the program and sensing means. (*Bottom*) Mechanically driven predetermining counter.





Photograph courtesy of Fairchild Hiller Industrial Products Division

Fig. 13–8. (*Top*) [illegible] by progressive acceleration and deceleration. (*Bottom*) Heavy-duty differential gear assembly with cover removed.

shown in the block diagram. The photo shows a heavy-duty differential gear assembly with the cover removed.

Advantages and Disadvantages of Clutch-Brake Systems

The principal advantage of clutch-brake systems is their versatility. Clutches and brakes can be obtained in many different sizes, varying from less than one inch to several feet in diameter; they are found in everything from instruments to heavy-duty applications. Both dwell and cycle times are easily programmed, making possible such things as very long dwell periods followed by very short indexing periods, operator controlled dwell periods, output motions of several hundreds (or thousands!) of revolutions following instantaneous or long dwell periods, etc.

Another advantage of the [illegible] that acceleration and deceleration can often be programmed or [illegible] in some clutch-brake systems. Hysteresis, eddy current, and magnetic clutches, for example, all produce output torques that are proportional in some way to the amount of [illegible] passed through the [illegible] coils. Some torque control is also possible with friction [illegible] clutches although these will wear rapidly if operated for long [illegible]

Clutches and brakes generally have much lower inertia than the prime mover in a typical situation; thus they can be used to produce much higher cycle rates than the designer could obtain by merely starting the prime mover on and off. This gives them an advantage over some other systems, though not over simpler intermittent mechanisms.

We have seen in earlier chapters that machine elements can be designed to be elastic [illegible] high-speed indexing applications. Dampers are often required to quiet undesirable [illegible] in a system following rapid index motions. However, in clutch-brake systems, "damping" in free [illegible] a brake is also an excellent damper!

One disadvantage of clutch-brake systems is their cost, although we must hasten to add that cost must always be related to performance, life, etc. Certainly a clutch-brake system costs more than a simple ratchet or Geneva drive, but it will also do a lot more. Clutches and brakes are apt to generate quite a lot of heat, depending, of course, on the amount of power they are transmitting and/or dissipating. The heat comes from electrical excitation currents, friction between slipping surfaces, eddy currents generated in conductive members of the clutch by excitation currents, and by repeated impact energy losses. These latter (eddy current and impact losses) contribute significantly to heating, only if cycle rates are high.

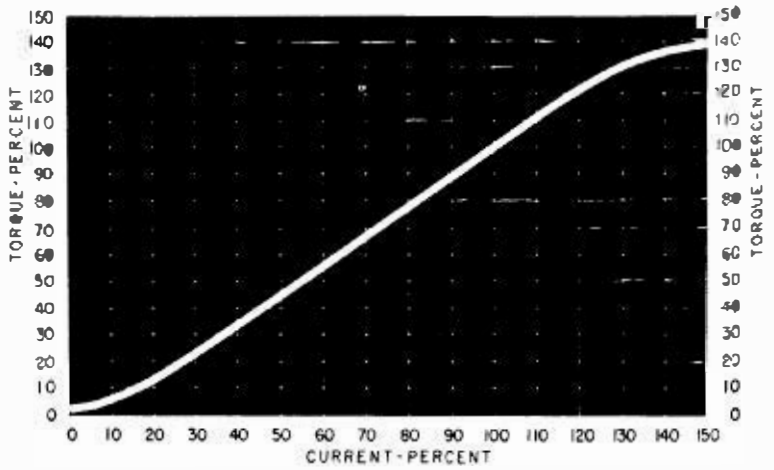
Finally there is the problem of control. The designer must arrange to sequence both clutch and brake properly. He does not want to energize both at once. At the least, this would result in a serious inefficiency in the system. In many cases it could also result in severe damage to the clutch and/or the brake. The designer does not want to have neither clutch nor brake energized at a given moment, because the load then would be completely uncontrolled. Instead, he must arrange to synchronize the operation of these two components correctly so that the brake takes over as the clutch releases the load, and vice versa. If he buys the clutches and brakes as separate units the entire burden of control is on

the designer. If he buys combined clutch-brake units the synchronizing will probably have been taken care of by the manufacturer.

Motion Curves

Figure 13–9 shows simulated motion curves for a toothed clutch-brake system. Motion starts with an impact as the clutch engages the load. It stops with an impact as the brake engages the load. Although the motion curve has been shown as a fairly straightforward one involving single impacts to start and stop the motion, in actual practice elasticity in the driving and driven members would result in multiple impacts at each end of the indexing cycle and the motion curve would probably be similar to that obtained for simple ratchet drives as discussed in Chapter 7. It is apparent, therefore, that toothed clutches and brakes can only be used with light loads and/or relatively slow speed systems.

Figure 13–10 shows the torque-versus-excitation current curve for a magnetic particle clutch. A clutch (or a brake) can impact a load only if it can apply torque instantaneously to the load. Since it always takes a finite period of time for the current in an excitation coil to reach drive values, this graph shows that a magnetic particle clutch or brake will always accelerate and decelerate a load gradually. Figure 13–11 shows the resulting motion curves for a magnetic powder clutch. Hysteresis, eddy current, and magnetic fluid clutches will have different but similar curves.



Courtesy Lear Siegler, Inc.

Fig. 13–10. Torque-versus-excitation current curve for a magnetic particle clutch.

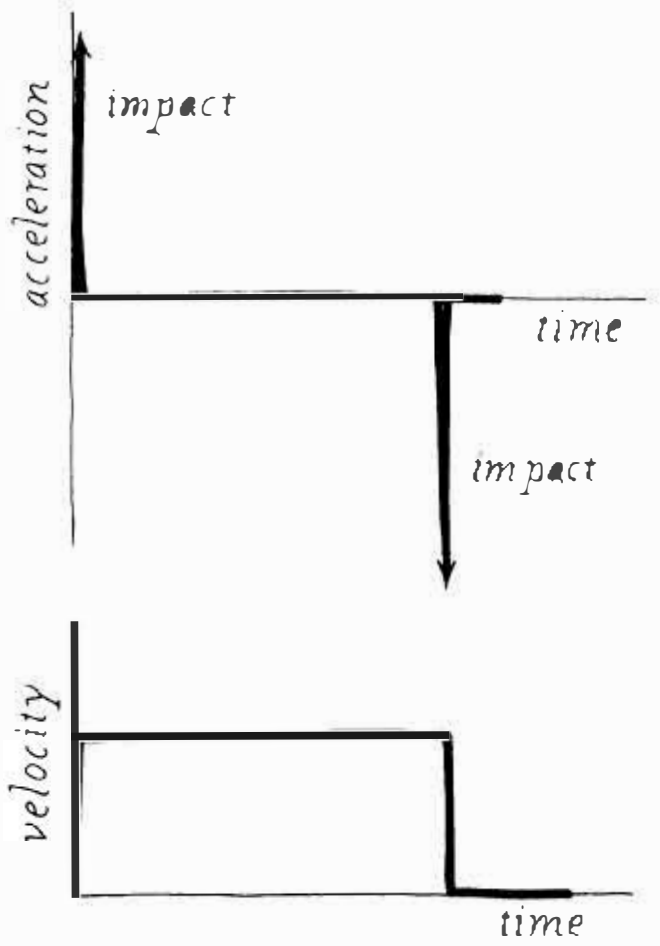


Fig. 13–9. Velocity and acceleration curves for a toothed clutch-brake system.

Friction disc clutches and those involving sprags, cams, rollers, etc., will have motion curves somewhere in-between that of the tooth clutch and that of the magnetic load particle clutch, since these types can pick up a load more rapidly than a magnetic clutch, but will not pick it up as rapidly as a solid tooth clutch would.

Clutch Brake Ratings

Different manufacturers unfortunately use different methods of rating their clutches and brakes. The designer will usually be concerned about response time, load capacity, heat generation, and control requirements, but he will not find all of these factors discussed in each manufacturer's literature. Some concentrate on one, some on another. Different systems of units are often used; with torque, for example, being given in grams-centimeters, ounce-inches, pound-inches, pound-feet, etc. The torque capacity of some designs is given in terms of the torque produced by a fully energized, fully engaged clutch or brake, while other manufacturers list the maximum dynamic torque which a clutch or brake can safely produce when suddenly energized at maximum operating speed. Some manufacturers do not use torque ratings at all, but instead, prefer to rate their equip-

Fig. 13-11. Acceleration, velocity, and displacement curves for a magnetic powder clutch.

ment in terms of the maximum "inertia" that the devices can handle at rated speeds. Unfortunately, they do not use actual inertia in slug-ft² or ft-lb-sec² but use instead, the electrical motor manufacturer's pseudo inertia of WK^2 with units of lb-ft², thus, if a designer is calculating true inertia in slug-ft², he must then divide his computed answer by 32 to convert it to lb-ft².

In the illustrations which follow later, I have attempted to list typical response times and dynamic load torques for the various designs shown. Such values, of course, are always suspect in a hardcover book, since manufacturers are continuously improving their products, therefore, these numbers are intended for general guidance only. (See manufacturers' catalogs for the latest values.)

Clutch and Brake Excitation

Although, as mentioned, clutches and brakes can be actuated by mechanical, pneumatic, or hydraulic means, most intermittent motion systems will use electrical excitation. Since proper excitation involves a little more than just closing a switch, this topic should be considered before looking at some specific clutch designs. This discussion is also pertinent for the designer who wishes to excite solenoid driven ratchets or escapements, stepping motors, etc.

In all electrical clutches and brakes with which we are concerned, the excitation current is passed through an electrical coil which produces an electromagnetic field. This field, in turn, acts on magnetic powders or fluids, generates eddy currents or hysteresis effects, or acts on ferro-magnetic clutch or brake members to pull them into or out of engagement with other elements in the devices. The more rapidly the current can be built up in the exciting coil, the more rapidly the various clutching and braking effects will occur. If the electrical coil were a purely resistive component in the circuit, then the current through the coil would respond instantaneously to any change in applied voltage across the coil. Coils, however, have inductance as well as resistance thus, they oppose rapid changes in current. Figure 13-12 shows the response of resistive and inductive-resistive loads to a sudden change in applied voltage.

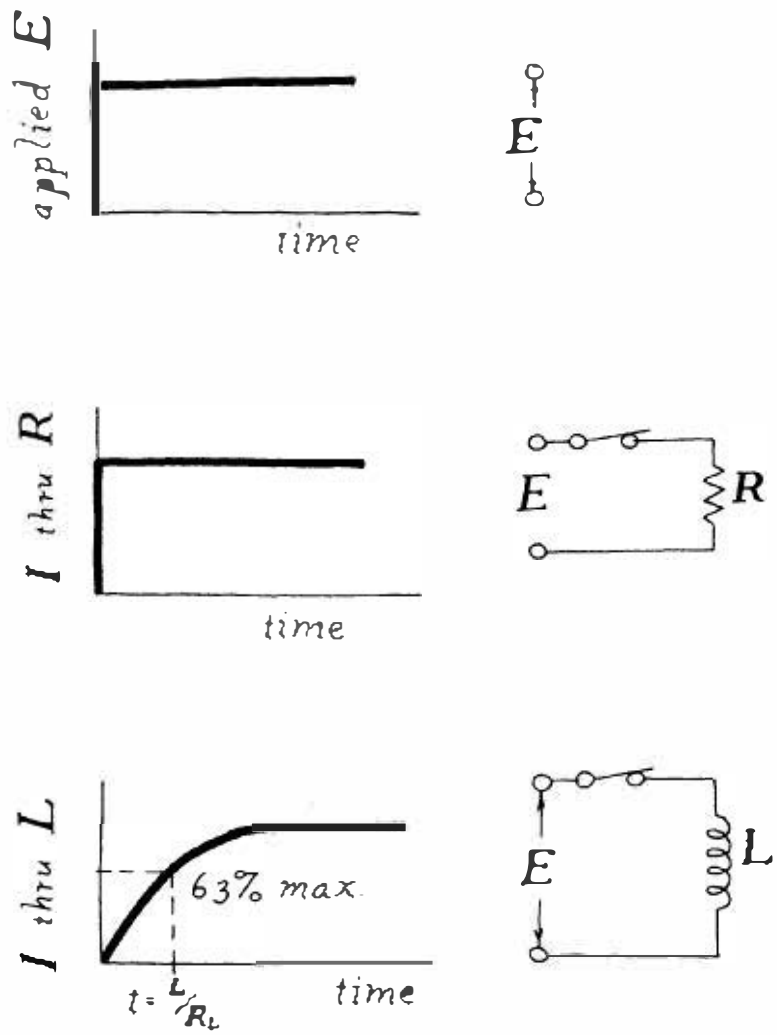


Fig. 13-12. Applied voltage and current curves for resistive and inductive circuits. Time L/R is called the time constant of the inductance and is the time required for current in the inductance to reach 63 percent of its maximum or steady-state value.

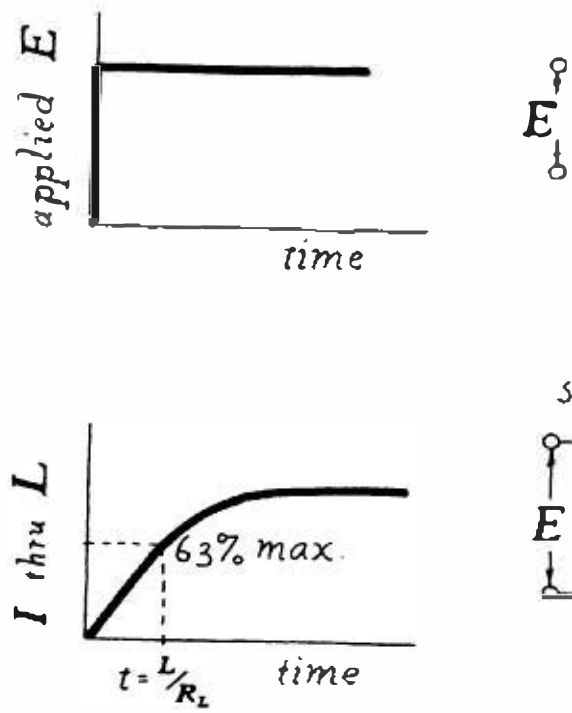


Fig. 13-13. Comparison of time constants for inductive circuits with and without forcing resistors.

Time Constant and Forcing Resistors

It is impossible to build a pure inductance. All coils have resistance as well as inductance. The ratio of inductance to resistance (L/R) is called the *time constant* of the coil current; the time required to reach 63 percent of its maximum or steady state value as shown in Fig. 13–12. The amount of electrical flux (ϕ—a measure of the strength of the magnetic field produced by the coil) will be directly and instantaneously proportional to the current in the coil. The more rapidly the current builds up, the more rapidly the magnetic field builds up and the more rapidly the clutch or brake will operate.

One way to increase the speed of operation is to reduce the time constant of the system by adding a resistance in series with the coil, as shown at the bottom of Fig. 13–13. This decreases the ratio of inductance to total resistance (L/R) in the circuit, and the current thus follows the applied voltage more rapidly. A disadvantage of this approach is that power is now wasted in the external resistance which is not doing any useful work. Nevertheless, an external resistor of this sort is called a *forcing resistor* and is often used if a fast response time is desired.

An inductance will resist the decay of current in a coil just as it resists a sudden increase in current. Figure 13–14 shows the way that current (and mag-

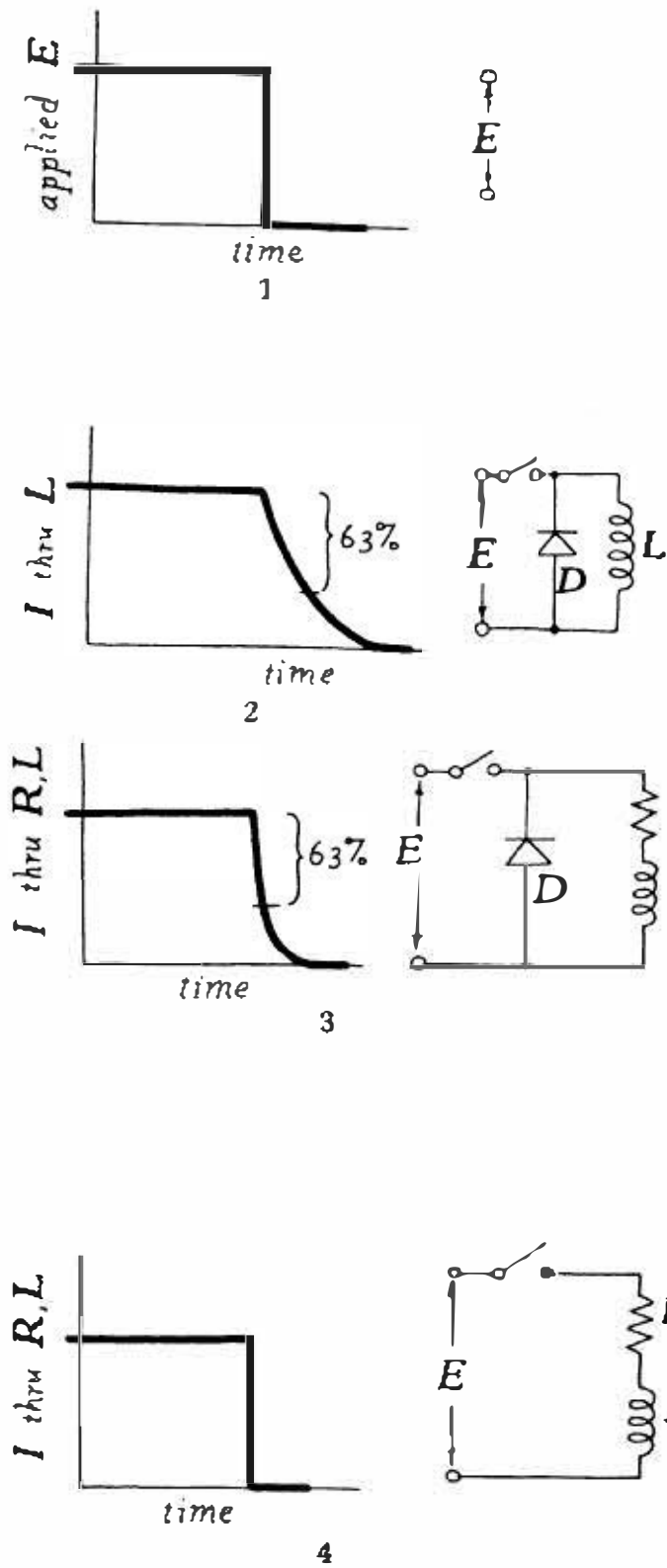
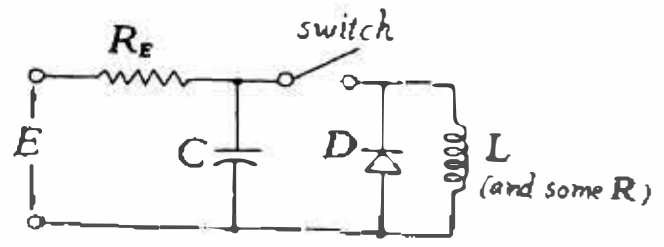


Fig. 13–14. Decay of current in an inductive circuit with and without forcing resistors. The curve in Part 4 shows what happens if an arc suppression diode (or substitute) is not included in the circuit.

netic flux) decay when voltage is removed from an exciting coil in a clutch or brake. Notice that the time constant now indicates the time required for the current to *decay* 63 percent from its initial value. Notice too, that an external forcing resistor speeds up this change as shown in diagram 3 of Fig. 13–14.

It should be noted also, that the current through an inductance will decay slowly after removal of the applied voltage only if there is some place for this current to go. In diagrams 2 and 3 of Fig. 13–14, for example, a diode, *D*, has been placed across the forcing resistor and the coil. In diagram 4 there is no diode and the current in the system stops suddenly with a voltage "back-spike" which will appear as an arc across the switch contacts that are controlling the circuit. The diode, *D*, in Fig. 13–14 is called a *suppression diode* because it acts to suppress this arc. Resistors and/or capacitors or neon lamps can also be used for arc suppression.



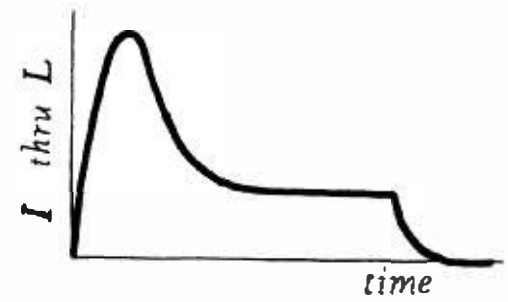
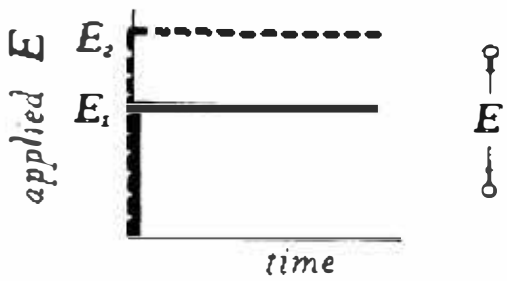
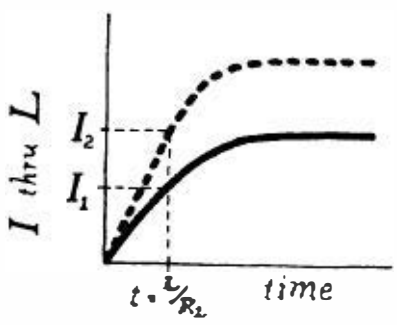


Fig. 13-16. A capacitor discharge drive circuit for an inductance. The graph shows the resulting current through the coil.

Over-voltage

Earlier, it was seen that the time constant is a measure of the time required for the current in the coil to reach 63 percent of the maximum, or steady-state value. This steady-state value, of course, is determined by the applied voltage. If a larger voltage is applied to the coil the steady-state value will be greater. The time constant will not be changed, but the current achieved by a certain time will be greater for the higher voltages, as seen in Fig. 13–15. Notice that the current through the coil achieves the value I_1 a lot sooner than time L/R when the higher voltage is applied to the coil. If I_1 is the current required to actuate the coil, therefore, the brake or clutch could be forced to act more rapidly if we applied a greater-than-necessary-to-produce-I_1-voltage, called *overvoltage*. The disadvantage, of course, is that the steady-state current through the coil would be a lot greater than that required to do the final job. This would waste power and/or produce excessive heat in the system. We can have our cake and eat it too, however, if the circuit shown in Fig. 13–16 is used. With the switch open, capacitor *C* will slowly charge through external resistor R_E until the voltage across the capacitor equals the applied voltage *E*, and current ceases to flow. When the switch is closed, the charge in the capacitor will be suddenly dumped through the coil *L*. We have suddenly applied voltage *E* to the coil. If the switch remained closed, however, the voltage across the coil will fall to some value less than *E*, because there will be a voltage drop across resistor R_E as long as current flows in the system. The resulting current through the coil will look something like that shown





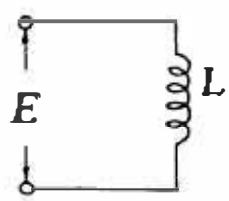
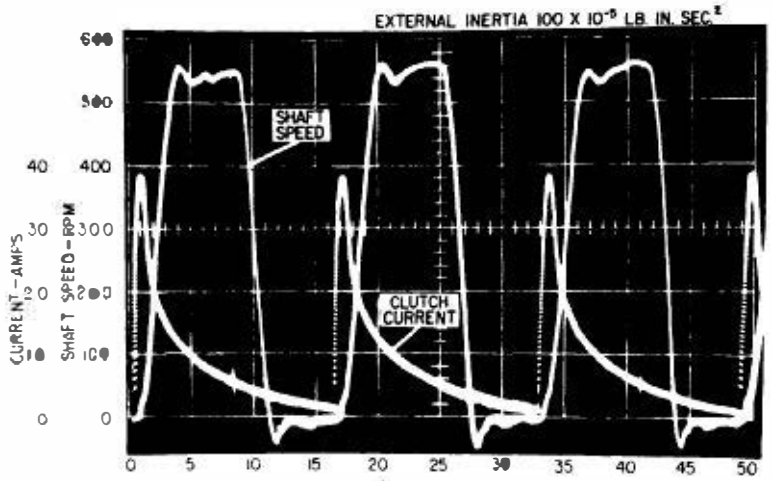


Fig. 13–15. The rate-of-rise of current in an inductance depends upon the applied voltage. In each case the current will reach 63 percent of steady-state value in time, $T = L/R$ but the steady-state value is increased by an increase in applied voltage.

in the curve at the bottom of Fig. 13–16. The peak on the left of the curve shows the current through the coil (from the charged capacitor) when the switch is first closed. This current then decays to a steady-state value supplied by source E and limited by R_E (mid-section of the curve). When the switch is re-opened, the coil current "decays" through the diode as shown by the right-hand end of the curve.

This is a very efficient system from an energy standpoint and it results in a rapid actuation of the clutch or brake with which it is used; but the maximum cycle rate or number of operations per minute, is usually reduced by this technique because time must be allowed for the capacitor to charge through



Courtesy of Lear Siegler, Inc., Power Equipment Div.

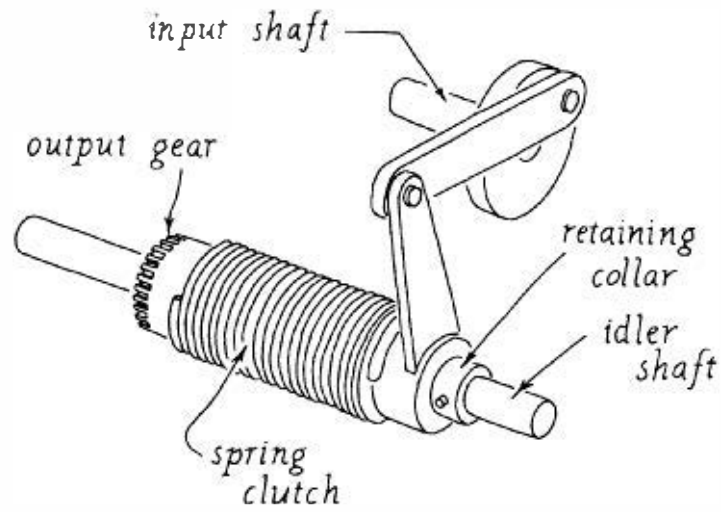
Fig. 13–17. Output shaft velocity and clutch current-versus-time for a magnetic particle clutch driven by a capacitor discharge circuit.

a relatively large resistor R_E, in order to take full advantage of the technique.

Figure 13–17 shows load velocity (shaft speed) and clutch current as a function of time for a magnetic particle clutch in which a capacitor drive (overvoltage) is used. A brake (whose current is not shown) prevents the output shaft from turning in-between clutch actuations. The clutch whose motion is described by these curves is operating at its maximum rate, so the clutch current is allowed to decay continuously and has no lower steady state value. The operation, however, is really the same as that shown in Fig. 13–16.

Examples

Now let us examine some clutch and brake systems. Figure 13–18 is a simple clutch system. It is a



Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 13–18. Simple spring clutch system.

possible embodiment of the general system shown in Fig. 13–1 where a reciprocating crank drives a one-way clutch to produce an intermittent output motion. In this case, the clutch is a spring clutch which tightens and produces output every time the input motion is in a direction to wrap the spring around the barrel shafts on which it is wound (as in Fig. 13–2 schematic, at right).

A simple ratchet clutch is shown in Fig. 13–19. This design is acceptable for slow-speed applications. This system was used on a monotype casting machine; input rotated at approximately one rpm and the input disc was about 8 inches in diameter. Friction and inertia served as the brake. (Compare this illustration with Fig. 13–2.)

Figure 13–20 shows another brakeless system which utilizes a pneumatic or hydraulic actuation of

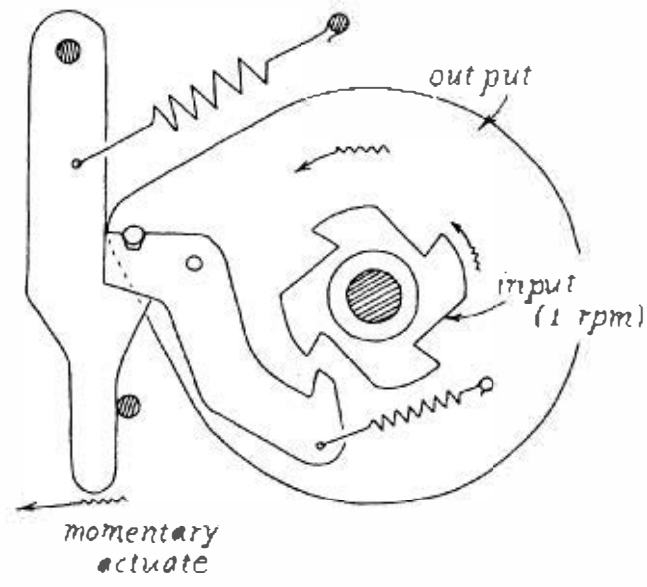
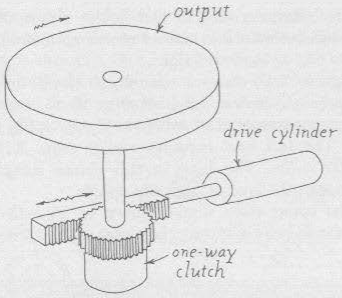
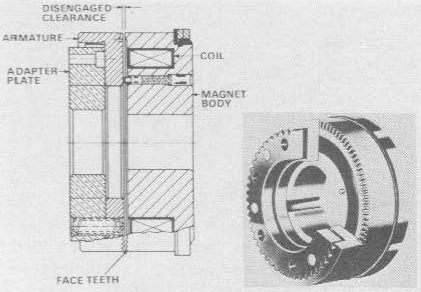


Fig. 13–19. Simple ratchet clutch.



Drawing courtesy of The Lowell Corporation

Fig. 13–20. Brakeless system with pneumatic or hydraulic actuation of a mechanical one-way clutch.



Courtesy of the Bendix Corporation

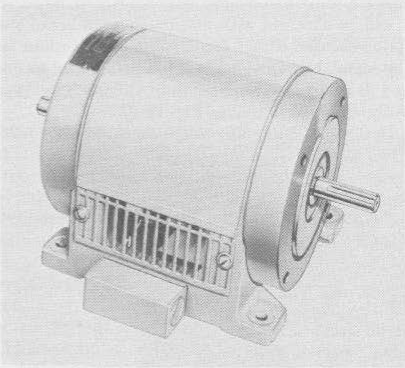
Fig. 13–22. Details of face-tooth clutches. (*Left*) Construction details. (*Right*) Cutaway view.

a mechanical one-way clutch. The clutch could be either a ratchet, sprag, cam, or roller clutch.

Construction details of an electrically actuated multiple-disc friction clutch, shown at left in Fig. 13–21. Typical torque capacity, depending on clutch size, ranges from 72 to 4,800 lb-inches. Cutaway view of this clutch is shown at right.

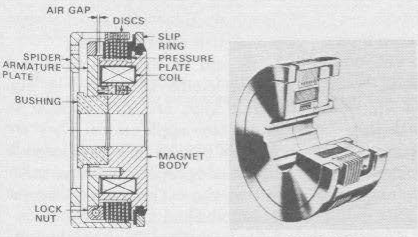
The face-tooth clutches in Fig. 13–22, can produce more torque for their size than any other type except, perhaps, a ratchet clutch which also uses solid teeth for the clutching member. Typical load capacity of this type of clutch ranges from 480 to 48,000 lb-inches.

The external view of a large-disc clutch brake is shown in Fig. 13–23. Its typical response time is 20 to 300 milliseconds and the torque capacity reaches 8,400 lb-inches. Compare this illustration with the friction-disc clutch brake in Fig. 13–24.



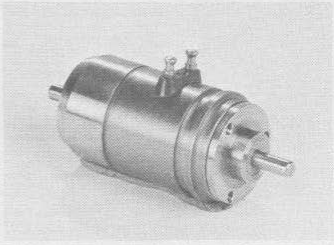
Courtesy of Eaton-Yale-Towne, Inc.

Fig. 13–23. Large-disc clutch brake.



Courtesy of the Bendix Corporation

Fig. 13–21. Electrically actuated multiple-disc friction clutches. (*Left*) Construction details. (*Right*) Cutaway view.



Courtesy of Magtrol, Inc.

Fig. 13–24. Friction-disc clutch brake for instrument applications.

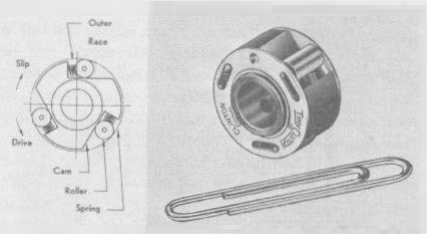


Fig. [illegible] ...nature rol... tch. (*Left*) ... details. (*Right*) Size compared with normal paperclip.

Figure 13–24 shows a friction-disc clutch brake intended for instrument applications. It is a spring-engaged brake and magnet-engaged clutch, however; it can be ... magnet-engaged brake and spring-engaged clutch, etc. Typical specifications for ... clutch brake are 0.2 to 6.5 lb-inches load torque and 28 ... ponse time. Compare ... ype of clutch brake to that in Fig. 13–23

In the ...ure roller clutch shown in Fig. 13–25, ... input ... tion inserted in ... sleeve which ...itute ... member. When ...he roller ass...y is r... the clockwise direction the rollers will be driven back against the small springs and the output will not be driven; it should be braked, however. When the direction of rotation of input is reversed, the rollers will wedge between the input member and the output sleeve producing rotation. If the output sleeve is sized properly, there will be little or no backlash.

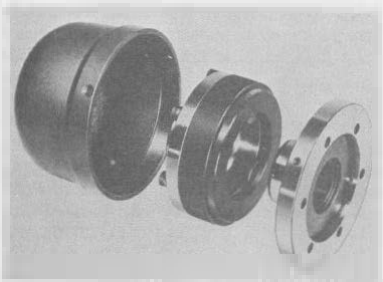
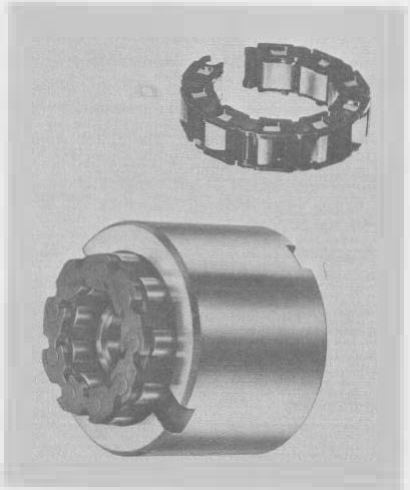


Fig. 1... Cam clutch with wedge cams used as gripping member.

Typical torque rating of this device ranges from 5 to 50 lb-inches. Note size of clutch compared with paperclips as shown at right

Figure 13–26 shows a cam clutch ... oper...ation to the roller clutch of Fig. 13-..., but with wedge cams instead of rollers as the ... member. The full load ...pacity ranges from 2,160 to 20,400 in-lbs. The cams in this clutch completely disengage the load at high slip speeds

The sprag ...utch shown in Fig. 13–27 is simpler in principle than the clutches shown in Figs



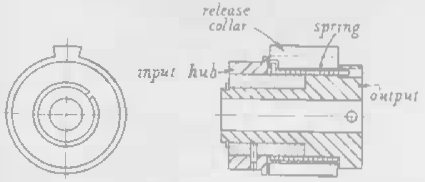
13–25 and 13–26, but it ope... ...h small wedges called sprags instead of ...th rollers or cams

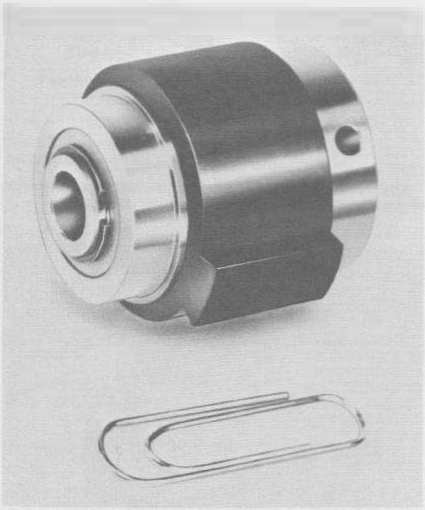
Construct... ... and an external view of a simple miniature ... clutch are shown in Fig. 13–28 Clutches of this type are very popular in high-speed business machine applications They are extremely compact as can be seen from the clutch in the photograph, and if made properly, have no backlash and will last a long time ... torque rat... from 15 to 6...

Two different designs are illustrated by the cut... a... ...ation of an electrically actuated spring clutch shown in Fig. 13–29. Series ... is normally

disengaged, and Series *EE* is normally engaged. Typical response time is one to 3 milliseconds; typical torque capacity, 30 to 450 lb-inches.

A new development, the one-way fiber clutch, using a linear oriented fiber material produced by the Minnesota Mining and Manufacturing Company,



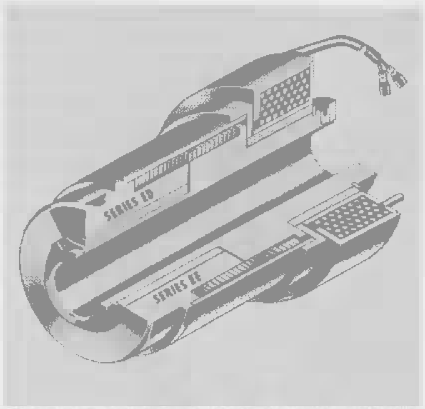


Courtesy of Helander Products Inc.

Fig. 13–28. Miniature spring clutch. (*Top*) Construction details. (*Bottom*) Size compared with normal paper clip.

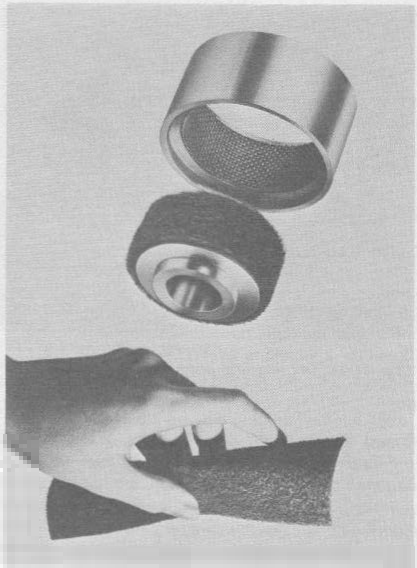
is shown in Fig. 13–30. The clutch will slip in one direction, but will drive in the other. Typical torque is 10 lbs. per square inch of friction surface or approximately 60 lb-inches, for a clutch that is two inches in diameter and one inch in length.

Figure 13–31 illustrates a dry magnetic powder clutch or brake. See Fig. 13–17 for a [illegible] curve of



Photograph courtesy of Marquette Metal Products Company

Fig. 13–29. Cutaway illustration of an electrically actuated spring clutch.

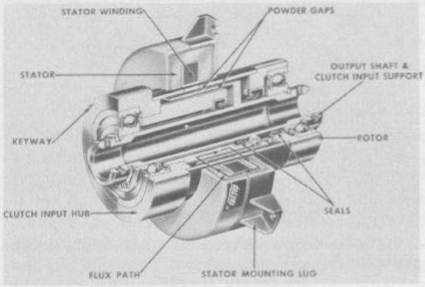


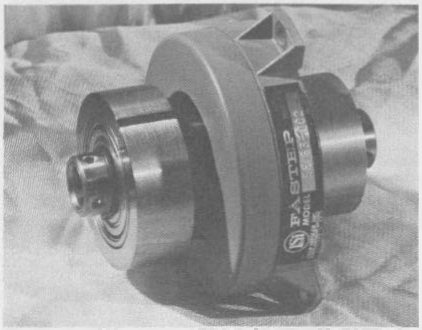
[illegible]

[illegible]

utch. The ty[illegible] esponse time i[illegible]3 to 40 milliseconds; typical torque capacit[illegible] 15 to 70 lb-

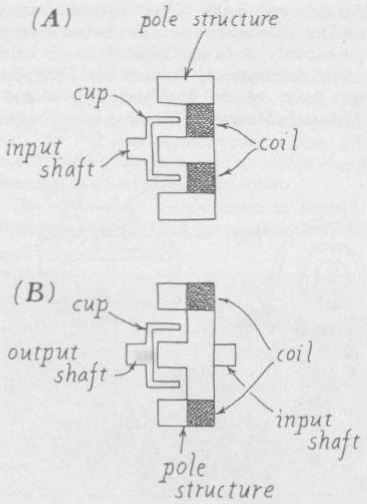
A highly simplified cross section of a hysteresis brake is shown [illegible]ig 13-32A. The pole structure containing the energizing coil, is grounded and does not rotate. The input shaft to be braked is keyed to a permanent magnet cup [illegible] rotates in a cup-



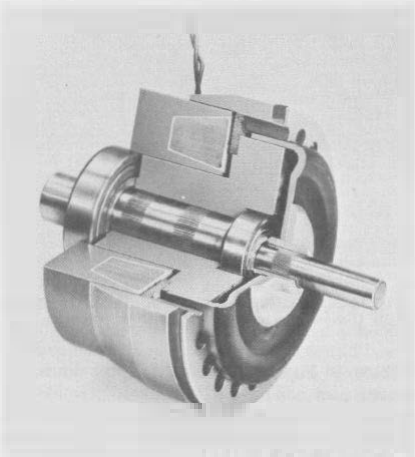


[illegible] details and cutaway view. (*Bottom*) [illegible] view

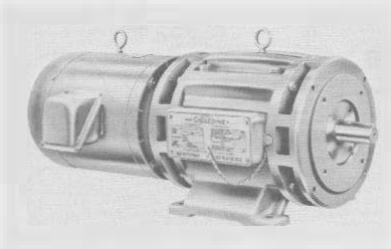
shaped air gap in the pole structure. When the coil is energized, magnetic flux linking cup and pole structure provides a braking action which is gentle or sudden, depending upon the amount of current passed through the coil. Figure 13-32B illustrates a hysteresis clutch which is nearly identical in construction to the hysteresis brake. In the clutch, however, the pole str[illegible]



[illegible] Fig. 13-32. Schematics showing the operating principle of [illegible] is brake (A), and the



[illegible] Fig. [illegible] Typical [illegible] hysteresis [illegible]

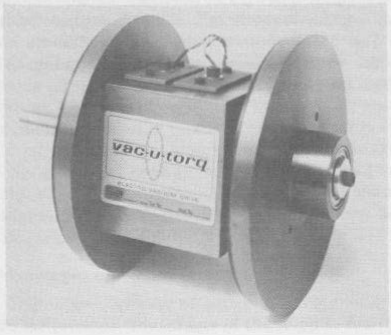


mounted on the output shaft. The magnitude of the coil current is varied to alter the amount of torque transmitted from input to output shafts.

Figure 13–33 shows a cutaway view of the hysteresis clutch shown schematically in Fig. 13–32. The construction of the hysteresis brake is nearly identical. The typical response time is 15 to 500 milliseconds; typical torque capacity, 0.125 to 100 lb-inches, depending on the clutch size.

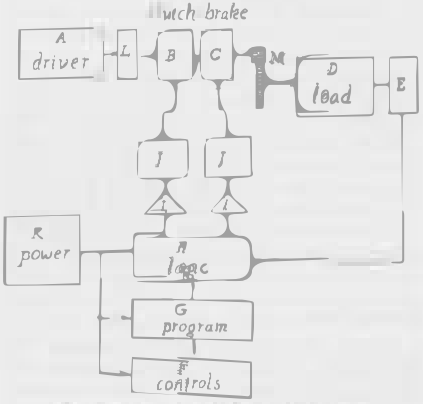
The manufacturer goes one step further in the combined motor-clutch-brake system shown in Fig. 13–34. An electric motor is provided with his clutch-brake making a complete indexing power plant when used with the control package also supplied. Typical pick-up torque of this system at 1,800 rpm, is 168 to 3,600 lb-inches, depending on the size.

The vacuum actuated friction-disc clutch brake shown in Fig. 13–35 has exceptional speed and excellent heat-dissipation characteristics because of the flow of air through the system. The typical response time [illegible] 0 lb-inch load.



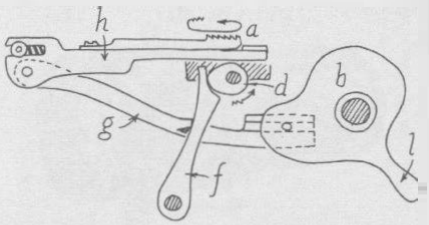
A complete clutch-brake system [illegible] shown diagrammatically in Figa 13–36. The elements of this [illegible]

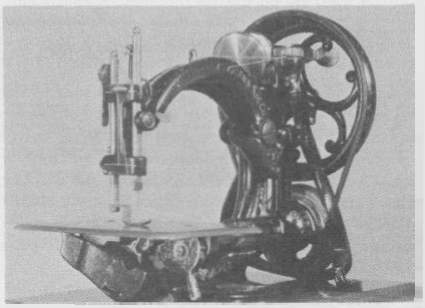
A. Driver—elec[illegible] motor or equivalent B [illegible]ically actuated clutch, C [illegible]ly actuated brake—preferably, clutch and brake ar[illegible] in one package so that one cannot operate without simultaneously releasing the



other; D Load, E. Sensor—an analogue, or more likely, a digital shaft encoder that pro[illegible] a trai[illegible] of h[illegible] pulses or a [illegible] of code "words" that lets the control system know where the load is; F. External controls—th[illegible] [illegible] simple [illegible], or more complex [illegible] panels [illegible] pr[illegible] basic control inter[illegible] with th[illegible] operator such as: "Start," "E[illegible]ency s[illegible]," "[illegible]wice," etc.; G. Pr[illegible] gra[illegible]er—necessary [illegible] there are expecte[illegible] to be frequent changes in such things as dwe[illegible] and [illegible] periods or distances, number [illegible] ste[illegible] per minute, [illegible]age levels, configu[illegible]ation [illegible] the [illegible] [illegible] [illegible]

into H in simple systems with inflexible control requirements; H Logic circuits—these convert external commands from F into electrical information for control of the clutch and brake. They may involve counting circuits, comparison circuits, etc.; I. Amplifiers—E, F, G and H operate at logic levels, perhaps 5 volts and milliamperes. The clutch-brake will need significant power so the logic outputs must be amplified at this point; J Forcing circuits—resistors, etc., to reduce the time constant of the clutch-brake to speed the response of the system to external commands, K. Power supply—must provide logic level outputs for E, F, G and H, and power levels for I. The former will have to be DC, the latter can be DC or AC depending upon the clutch-brake requirements. For the fastest response power, outputs must be DC also; L Fan—if the clutch and brake are operated with heavy loads and/or at high cycling rates some method of cooling them will have to be provided. M. Reduction gearing. The drive motor will usually run a great deal faster than the load, thus reduction gearing is required. The clutch-brake should be mounted on the fastest shaft to reduce torque requirements and inertia problems.





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machine feed mechanism. (*Bottom*) Another W[illcox] machine with similar feed

Intermittent Motion

Clutches are frequently used in production systems to intermittently feed of material or assemblies past a work station. As such, the [mechanisms] probably are distant descendants of the cloth feed mechanism in early sewing machines. In the drawing of the feed mechanism used in a [Wil]cox and Gibbs sewing machine, Fig. 13-37, the small cam (g) rotates continuously, acting through levers (f) and (y) to move lever (h) back and forth. The cam also periodically moves the feed teeth (a), which are mounted on (h). As a result, teeth (a) are moved in a roughly square path to advance the cloth through the machine intermittently. The photograph below (Fig. [13-37]) shows another unique Willcox and Gibbs sewing machine with a similar mechanism.

CHAPTER 14

Stepping Motors

Although not truly "mechanisms," stepping motors deserve consideration in this book because they comprise one of the most important of all modern-day methods for producing high-speed intermittent motion. They are electrical motors that are designed to take a single step when fed a single electrical pulse. Some types of stepping motors, or *steppers*, as they are often called, will also run continuously just as a "regular" motor if the input frequency is high enough; but most steppers are not used in this way.

Stepping motors are much in favor today since a large number of industrial and commercial functions are now controlled by digital computers or other digital equipment, and the stepping motor offers an excellent method of converting electrical pulses into various types of motion. Of course, such things as solenoid operated ratchets, clutch and brake systems, inverse escapements, etc., can also serve the purpose and are used to produce motion from pulses. The step motor, however, can operate at much higher speeds (thousands of steps per second in some cases) and with a much longer life than an impacting mechanism such as a ratchet. One of the most significant advantages of most stepping motors, in fact, is the absence of contact impact between input and output members. Accelerations and decelerations are comparatively gentle, extending the life of any machine which is driven by the device (see Fig. 5–6).

Advantages and Disadvantages

High speed, absence of impact and very long life are the principal advantages of the stepper. Another advantage, in some applications, is its versatility. Dwell and motion periods can be as long as desired (if the motion period includes one, or more than one, step). Steppers are frequently used, therefore, in systems where the dwell period or length of motion is expected to be variable; in digital process control systems, for example, where they might be used to position valves, etc.

Stepping motors are also very compact drive systems (compared to motor-clutch-brake combinations, for example). We tend to overlook the fact that a cam or Geneva must be driven by something. The cam looks small and compact, but the total intermittent motion drive system also involves a motor. With a stepper the "motor" is the whole system (except for the control circuits, of which some type is also required with "regular" motors).

Stepping motors are generally used in an open loop control system. There is no feedback since none is required, presumably, to position the load. If the stepper is fed five pulses it will move to a new position five steps beyond the first. Some designers, however, insist that it is very dangerous to count on this if your personal safety or the safety of your machine depends upon the certainty that the motor has obeyed you. Digital pulses have a way of getting lost or of appearing when least expected (thanks to electrical "noise").

Many designers, therefore, provide the stepper system with a feedback of some type; perhaps a shaft encoder on the load that is checked by the digital computer which has given the original instructions

to the stepper. Even then some designers still prefer an analogue system (servo loop) to a digital system because of the ever present "noise" problem.

The stepper is usually a very elastic drive since in most types there is no mechanical contact between rotor and stator. Forces are generated electromagnetically; this eliminates impact and makes possible a very long life, but it also means that the torque applied to the output shaft is elastic. This is a "springy" drive, and care must be taken to control hunting and vibration as a result. Dampers or brakes must be used in many situations.

Step motors also need rather complex drive circuits since most of them require two or more separate trains of input pulses and these various pulse trains must be properly phased with relationship to each other if the motor is to operate correctly. For the highest speeds under the highest loads, furthermore, there is an acceleration and deceleration phase during which the pulse repetition rate must be lower than the maximum that the motor can follow. This requirement further complicates the drive circuits. Finally, there is frequently the need to modify the drive circuit to "force" the motor coils and/or to energize the coils in a pattern that will increase the locking or holding torque, reduce vibration or hunting, etc.

As a result of all this, step motor drive systems can be quite expensive compared to the drive circuits for such mechanisms as ratchets and inverse escapements; but, of course, neither of these latter devices have the versatility of the stepping motor, hence a pure cost comparison is really not fair.

Magnetic Circuits—Some Basic Principles

Most engineers have not had much exposure to magnetic circuits (as opposed to electrical circuits), and it would probably be helpful to take a look at some basic principles before studying the specific configurations of various types of stepping motors.

Figure 14–1 shows a simple magnetic circuit and its electrical equivalent. An electrical coil is wrapped around one leg of a magnetic circuit which is comprised of three pieces of iron and an "air gap." There are also very small "air gaps" where the various pieces of iron are joined together. When the coil is energized, perhaps by placing a battery across its terminals, the coil will generate magnetic flux. This flux will seek the path of least resistance from one end of the coil to the other, and so will "flow" in the pieces of iron that comprise the magnetic circuit,

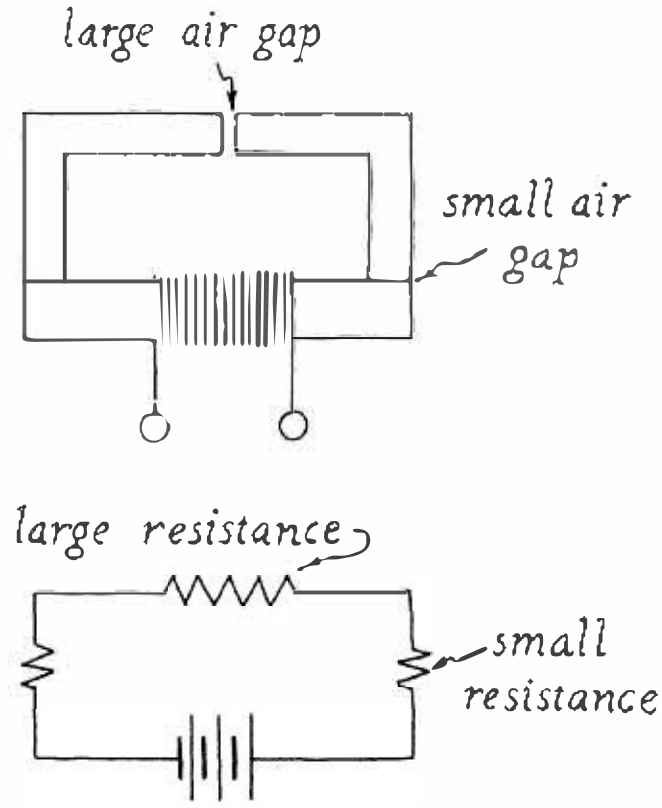


Fig. 14–1. Simple magnetic circuit and equivalent DC electrical circuit. A coil energized with direct current acts in a way that is analogous to an electrical battery; the first producing flux, the second, current.

just as electrical current will flow in a copper wire that connects the two terminals of a battery in an electrical circuit. In each case the metal is a lower resistance path than the surrounding air. To be rigorously accurate, we must recognize that magnetic flux will not confine itself completely to the iron but will also travel, to some extent, outside the iron (leakage flux). We can safely ignore this in the analyses which follow, however. In the magnetic circuit above, the flux is also forced to travel entirely in air for part of the way, through the three "air gaps." Air gaps reduce the amount of flux a given magnet can produce in a magnetic circuit, just as a resistor limits electrical current. The large air gap will have quite a bit of resistance to the buildup of flux. The small air gaps will also resist it, but to a lesser extent. This resistance to the development of flux is called *reluctance* in a magnetic circuit.

The direct current electrical circuit at the bottom of Fig. 14–1 is an exact analogue of the magnetic circuit. A battery generates current which passes through the circuit encountering one large and two small resistors. The larger the total circuit resistance, the more difficult it is to generate current in any part of the circuit. By the same token, the larger the air gaps in the magnetic circuit, the more difficult it is

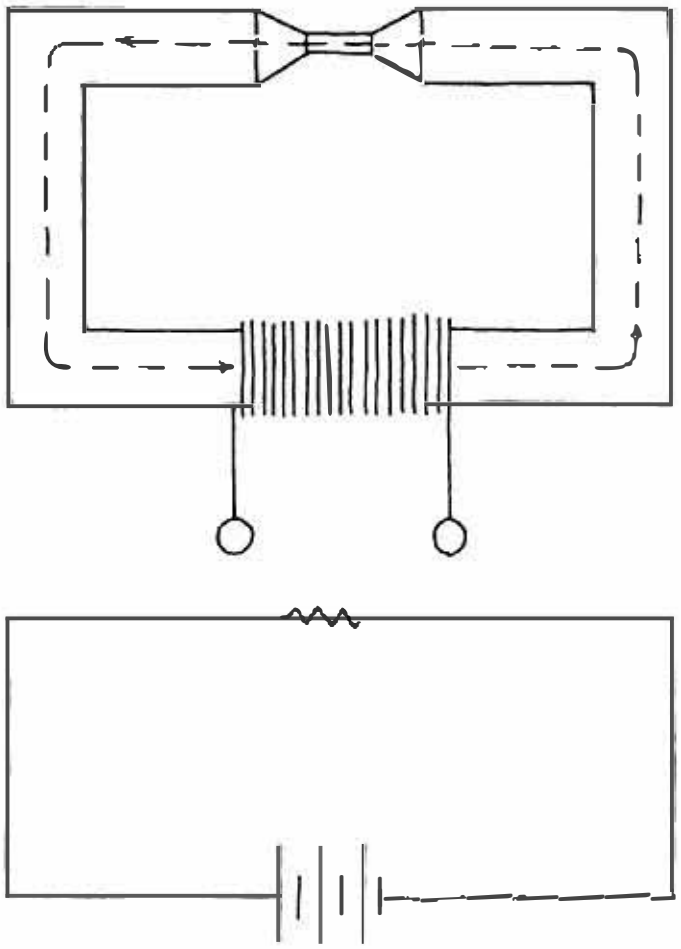


Fig. 14–2. Magnetic circuit containing one piece of iron with a reduced cross section, and equivalent DC circuit.

to generate magnetic flux in the iron and the air gaps.

Figure 14–2 shows another magnetic circuit and its electrical analogue. In this case, there are no air gaps, but part of the iron has been reduced in cross section. Since the amount of flux that can pass through a piece of iron depends, in part, on the iron's cross-sectional area, the reduced section also acts as a small resistor.

In Fig. 14–3, we see what happens when an attempt is made to drive a lot more flux through the circuit of Fig. 14–2. The reduced cross section, which had previously acted as a relatively small resistor, now saturates with flux until it can hold no more. Attempts to increase the flux still further reveal that the reduced cross section is now acting as an air gap of the same length. In effect, the reduced cross section has been taken out of the iron circuit. Unlike most electrical resistors, therefore, magnetic circuit reluctances are not constant values. They change in value as the flux density changes.

A parallel magnetic circuit and its electrical equivalent is shown in upper diagram of Fig. 14–4. The flux generated by the electrical coil now divides between two branches of the magnetic circuit just as the electrical current in the circuit (lower diagram) divides between the two legs of the electrical circuit. In each case, the amount of current (or flux) is dependent upon the amount of resistance (or reluctance) in the circuit. The left-hand leg of the electrical circuit carries more current than the right-hand leg because resistor r_2 is a lot bigger than resistor r_1. By the same token, the amount of flux in the right-hand leg is less, because air gap g_1 in the magnetic circuit has a lower reluctance than air gap g_3. This is because the pieces of iron facing each other at g_3 are not aligned, thus decreasing the area by which the pole-faces overlap. Less area means higher reluctance (just as an increase in air-gap length means higher reluctance) and, therefore, less flux. Misalignment of this sort is always encountered in stepping motors and it is useful to know how the flux will distribute itself in this situation.

Figure 14–5 (*Left*) shows another magnetic circuit that is similar in appearance to that shown in Fig. 14–4. In this case, however, all poles are aligned.

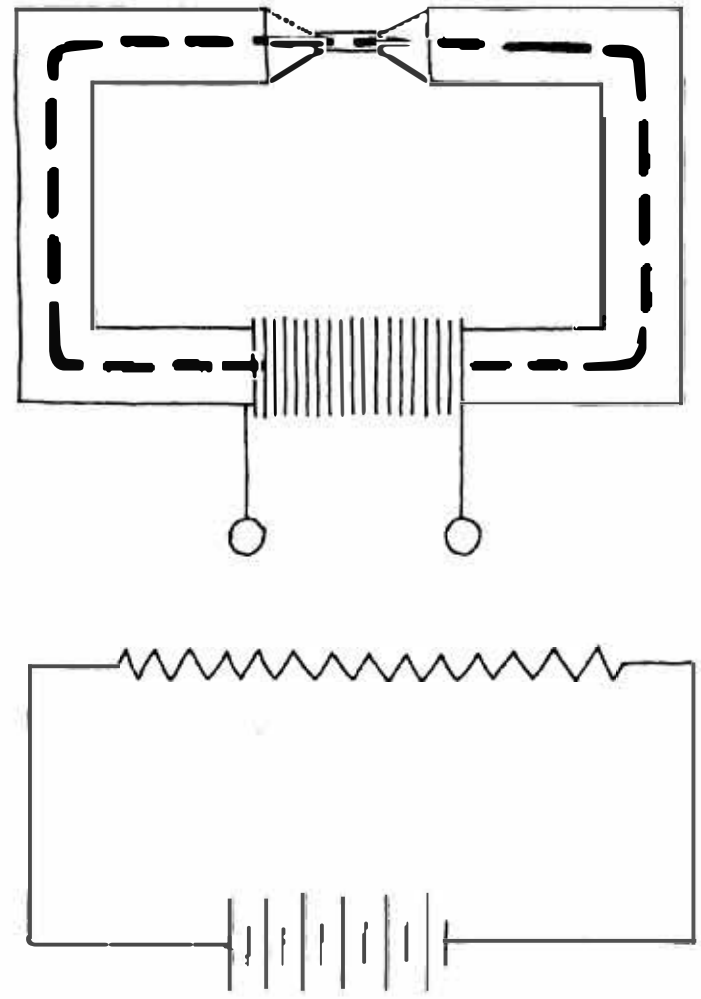


Fig. 14–3. Same magnetic and electrical circuits as in Fig. 14–2. With more current flowing in the coil more flux is generated.

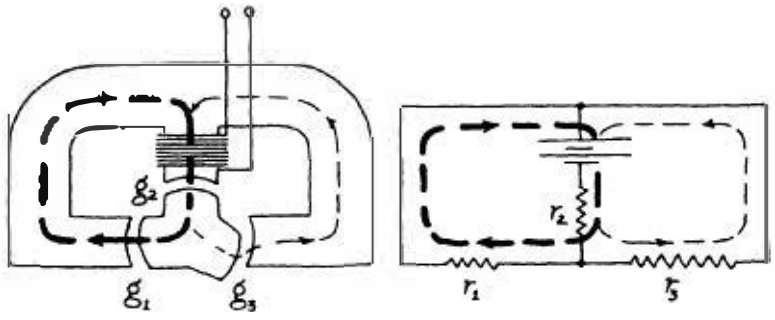


Fig. 14–4. Parallel magnetic circuit and its electrical equivalent.

Here, also, are three separate sources of magnetic flux. Two permanent magnets m_1 and m_2, and the electrical coil at the bottom. The permanent magnets are placed with their north poles on top, as shown. The electrical circuit equivalent for this magnetic circuit is shown in 14–5 (*Right*). At least this is the equivalent circuit as long as the coil at the bottom of the magnetic circuit is not energized. Under these circumstances there are only two flux producers, the two permanent magnets, and the iron loop containing the coil is, in effect, a short circuit between the two south poles of the two magnets. If these magnets are of equal strength, however, no flux will flow in this leg of the circuit; just as in the electrical equivalent circuit, no current will flow in the bottom wire as long as the two small batteries are equal.

Figure 14–6 (*Left*) shows what will happen when the coil in Fig. 14–5 is energized; with a north pole to the left, and a south pole to the right. The flux developed by the electrical coil will now aid the flux produced by magnet m_1, and oppose the flux produced by magnet m_2, producing heavy flux through some portions of the circuit, as shown with a heavy dotted line, and light flux in the other legs. In the electrical equivalent circuit (*Right*) a new battery, b_3 has been added to the circuit and the current pattern is then the same as the flux pattern of the magnetic circuit.

The polarity of the electrical coil has been reversed in Fig. 14-7 so that the north pole is now on the right-hand side rather than the left. The electromagnetic is now aiding permanent magnet m_2, rather than m_1 (*Left*). Reversing the polarity of battery b_3, in the equivalent electrical circuit (*Right*), accomplishes the same thing. Notice that in both cases (Figs. 14–6 and 14–7) there is very little flux in one of the two air gaps g_2 or g_3, whereas originally (Fig. 14–5), there was equal flux in both of the gaps.

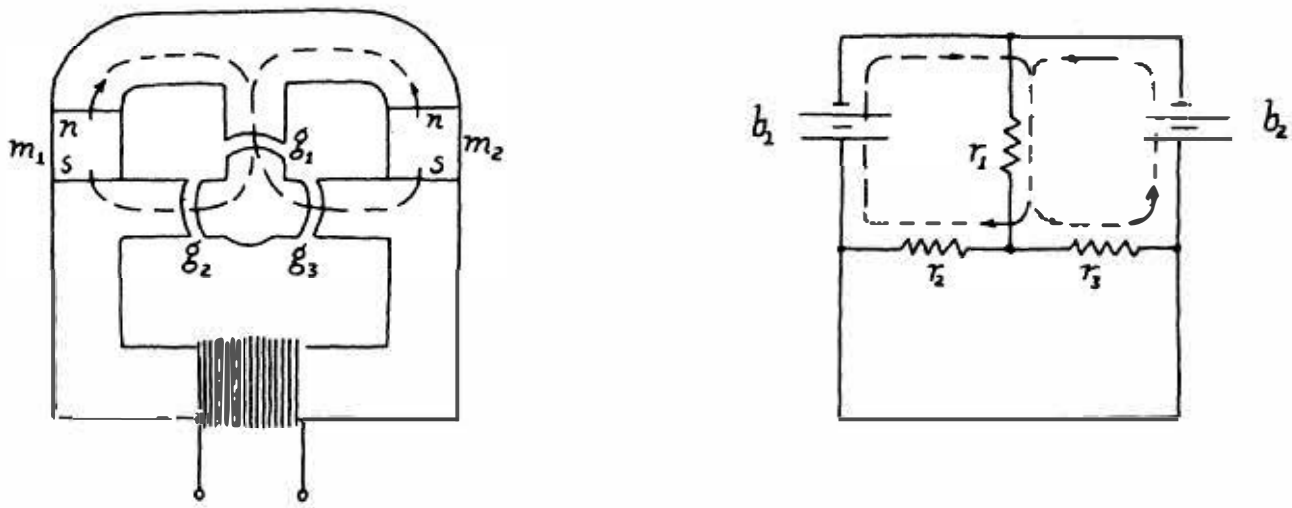


Fig. 14–5. Parallel magnetic circuit with permanent magnet and electromagnetic flux generators. (*Left*) In this illustration the electrical coil is not energized. Flux is produced only by the two permanent magnets as shown by the light dotted lines. (*Right*) Equivalent electrical circuit.

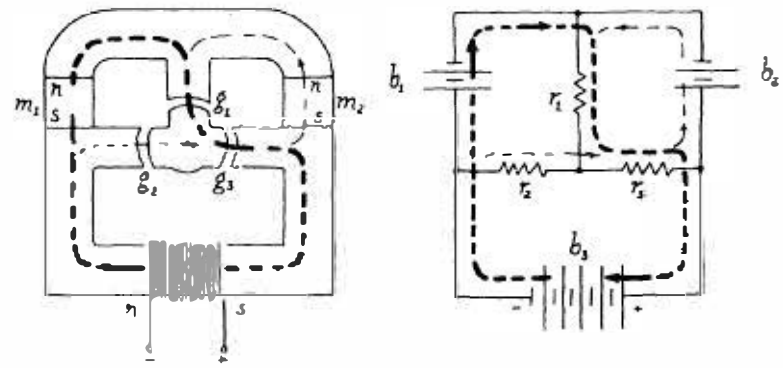


Fig. 14–6. The magnetic circuits of Fig. 14–5 with the electrical coil energized (*Left*). This coil acts to switch the flux of the permanent magnets as shown, producing heavy flux (heavy dotted line), and light flux (lighter dotted line). The electrical equivalent circuit (*Right*), behaves in the same fashion if a third battery b_3, is introduced.

Energizing the electrical coil has switched the flux so that most of it passes through one gap or the other. Flux switching of this type is encountered in many stepping motor designs.

Flux and Torque

We have seen that the amount of flux passing through a given leg of a magnetic circuit can be influenced by either air-gap length, air-gap area, or by flux-switching techniques. Now let us relate flux patterns to drive torque. In Fig. 14–8 two different electromagnetic circuits are seen. Circuit 14–8 (*Top*), is the same as the simple circuit of Fig. 14–1, but a rotatable piece of iron (hereafter called a rotor) has been inserted in the air gap. In the diagram, this rotor is at an angle to the adjacent legs of the magnetic circuit, reducing the area of overlap of the pole faces (rotor and circuit). As we have seen, a reduction in overlap area means an increase in the reluctance of the air gap. If the rotor is free to turn it will move to increase the overlap area, to reduce air-gap reluctance. And the system would do work while it turned. In other words, a weight could be hung from a pulley mounted on the rotor and the rotor would lift the weight as it moved to align its pole faces with those of the adjacent circuit.

Some people like to think of flux lines as rubber bands. Stretching a rubber band requires work. The band will do work as it "shrinks" in size. This is a useful and reasonably correct analogy for understanding the behavior of stepping motors and other magnetic circuits.

With a rotor in a magnetic circuit, torque is required to crowd flux lines through a smaller area or to increase the length of flux lines. In the circuit of Fig. 14–8 (*Top*) we could think of the "rubber band" flux lines as following the path sketched by the heavy dotted line. As the rotor turns, the dotted line shrinks in length, delivering energy to the system which provides work to move a load.

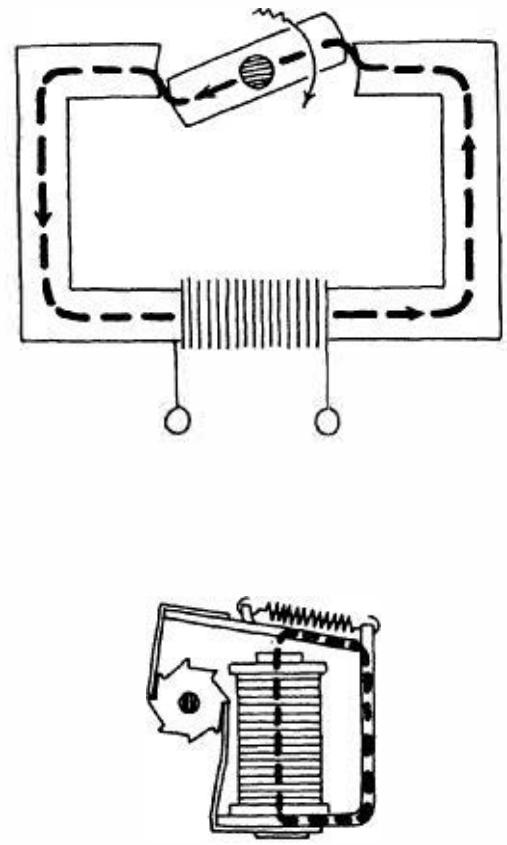


Fig. 14–8. Two different magnetic circuits. (*Top*) Simple magnetic circuit with rotor, showing stretching of flux in air gaps. (*Bottom*) Conventional ratchet motor with ratchet wheel driven as flux lines decrease in length.

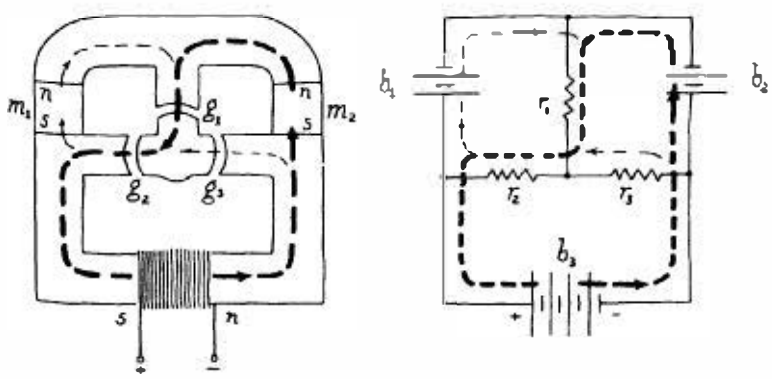


Fig. 14–7. The magnetic and electrical circuits of Fig. 14–6 with the polarity of the electrical coil (and battery b_3), reversed. (*Left*) Flux is switched in the other direction to that shown in Fig. 14–6. (*Right*) Current is also reversed.

Other electromagnetic power plants work on this "shrinking flux" principle. Something moves to shorten flux lines and the output work is taken from the thing which moves. In Fig. 14–8 (*Bottom*), as a second example, we see a conventional ratchet motor. A solenoid attracts a clapper whenever the solenoid is energized. The clapper, in turn, drives a spring blade which indexes a ratchet wheel. In this case, the "elastic band" flux lines are fairly tidy to begin with, but they do shorten as the device produces work to drive the ratchet wheel and its load. This is a more accurate rubber-band analogy than the first example, since the change in reluctance that is producing the work is a change in air-gap length rather than area. But the analogy is useful for both situations.

Stepping Motor Design

Consider now a magnetic circuit configuration that more nearly resembles a stepping motor. Figure 14–9

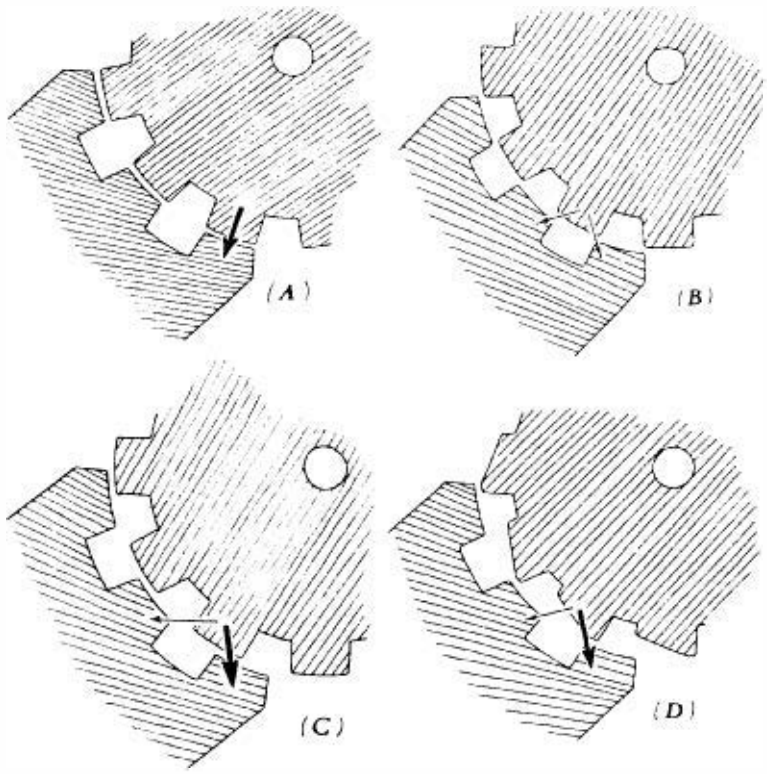


Fig. 14–9. Portions of a magnetic rotor and stator pole showing flux and torque produced in various stepping motor situations.

shows portions of a stator pole and rotor from a hypothetical stepping motor. Various possible alignments between these two members are shown in the four illustrations. Arrows indicate the intensity and main direction of magnetic force between rotor and stator in each case. A heavier arrow indicates more force than a light arrow. Although arrows are shown at only one pair of poles, force would also exist in equal measure and in the same relative direction between the other pole pairs.

In Fig. 14–9A, rotor and stator poles are perfectly aligned. This is the condition of minimum air gap and, therefore, of minimum reluctance. A large amount of flux will pass from rotor to stator, but because the poles are aligned, there is no "elastic band" to straighten at this point and no torque is generated between rotor and stator. If we attempt to move the rotor, however, a torque resisting such motion would be encountered as soon as there were any pole misalignment.

In Fig. 14–9B the rotor poles are positioned exactly halfway between the stator poles. The flux will now divide in half as it passes from rotor to stator. The air gap has increased and air-gap reluctance is high.

Again, no torque will be developed here if the rotor is positioned exactly midway between stator poles, since two stator poles will "pull" equally on each rotor pole.

In Fig. 14–9C, the rotor has been turned clockwise so that the bottom pole is more nearly aligned with one stator pole than with the other. There is partial overlap between the marked rotor pole and the middle stator pole so the amount of flux that passes from the marked rotor pole to the stator is greater towards one stator pole than to the other; just as in Fig. 14–4 the amount of flux passing through gap g_1 was greater than that passing through g_2. Under these conditions, there will be a net torque on the rotor in a counterclockwise direction. As the rotor moves, the amount of flux passing into the right-hand stator pole will increase further, and that into the left-hand stator pole will decrease until the rotor is again in perfect alignment, as in Fig. 14-9A. As was pointed out earlier, there would then be no drive torque on the rotor. Nor is there any drive torque in the situation of Fig. 14–9B, since the rotor is being pulled equally in two directions. Optimum drive torque occurs somewhere between these two conditions; perhaps in that position shown in Fig. 14–9C, where there is still some residual flux to the center pole of the stator tending to retard motion, but there is a large amount of flux into the right-hand stator pole operating with a good overlap to urge the rotor forward.

Figure 14–9D shows another way to produce a net torque on the rotor. The rotor teeth, or poles, have been modified to increase the air-gap length on one side of each pole and decrease it on the other. Notice that the rotor teeth in this case are positioned exactly midway between the stator teeth, as in Fig. 14–9B, but there is a net torque on the rotor in Fig. 14–9D because of the way that the rotor poles have been shaped.

Having explained flux patterns and torque between rotor and stator poles in a typical stepping motor, we should ask what is required to produce intermittent motion? First, there must be a torque to start a load in motion. The pole configuration of Fig. 14–9C would do that. A fraction of a second later, however, the output motion must be stopped. The rotor of Fig. 14–9C will turn until the poles are perfectly aligned, as in Fig. 14–9A. It will not stop there, however, because at that point no torque is developed between rotor and stator (and because, until that point, all torque which was developed tended to move the rotor in a counterclockwise direction). As the inertia of the rotor moves it past the perfect alignment point, however, a clockwise torque is produced, as the stator pole is being "left

behind." When the two poles are misaligned by about ¼ of a pitch (distance between two stator poles) the force pattern is a mirror image of that shown in Fig. 14–9C; the flux passes mainly through one of two stator poles but is now directed so as to produce a clockwise torque.

At some point between poles, therefore, the rotor will come to rest and reverse, moving in a clockwise direction until it again passes the perfect alignment position, again stops, again reverses direction, etc. It will usually do this several times, "hunting" for the perfect alignment position. Each swing carries it through a shorter arc since friction in the load and bearings tends to damp and eliminate this oscillation at about the rest or zero torque position. Once the rotor has come to rest it should be held there by some kind of detent action. This can be accomplished by permanent magnets or by keeping the drive coils energized. This detent torque must be overcome when the rotor moves again (usually when some other pole pair has been energized) thus it is desirable to minimize the detent torque when another step is taken. Electro-magnets, therefore, which can be turned off, are better (though more expensive) detent flux sources than permanent magnets. Detent torque can also be "switched" as in Fig. 14–6, etc.

The drive sequence described above is a simple one that is used in many stepping motor situations. It is possible, however, to energize "non-operating coils" of a stepping motor to help brake and detent a rotor. This is done fairly often since most stepper designers complain that it is easier to start a rotor (and load) than to stop it.

Types of Stepping Motors—Permanent Magnet Motors

One of the more popular types of stepping motors, the permanent magnet motor, is shown in Fig. 14–10 in a schematic illustration. The rotor is a permanent magnet. The stator consists of a magnetically soft iron structure and a number of energizing coils. In the illustration none of the coils are energized and the rotor is merely attracted (by its desire for a minimum reluctance flux path) to the nearest stator poles, those marked *a* and *e*. In Fig. 14–11 coil *b* has been energized to produce a south pole and coil *f* has been energized to produce a north pole (at the ends of each coil nearest the rotor). This will produce the magnetic flux shown by the heavy dotted line. Considering the flux as an elastic band we

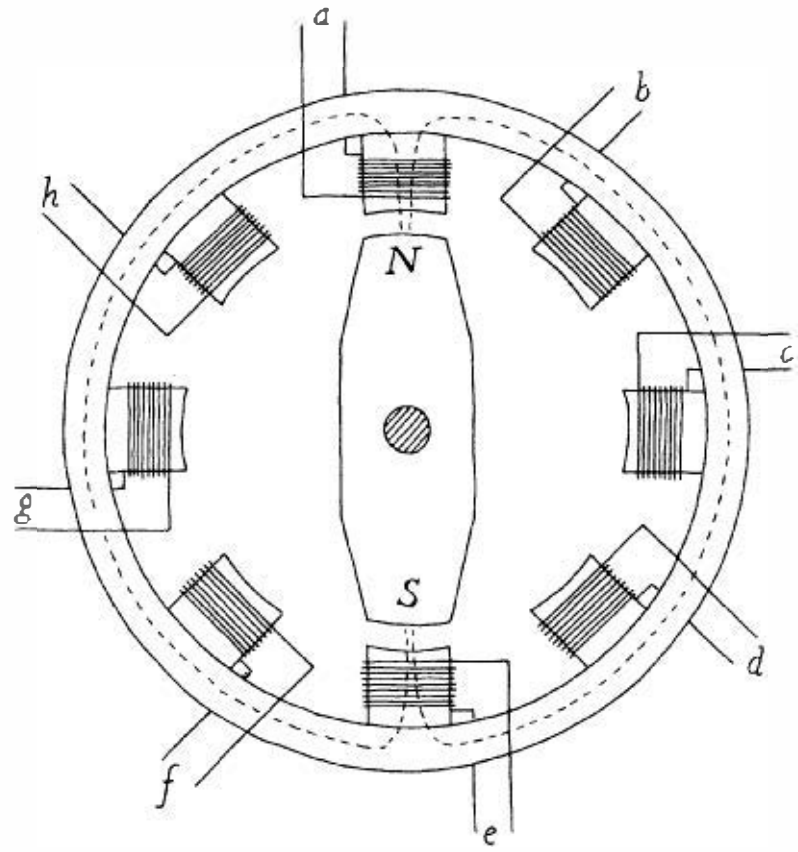


Fig. 14–10. Schematic illustration of a typical permanent magnet stepping motor. The rotor (marked *NS*) is a permanent magnet.

realize that the rotor must turn to shorten the flux lines between rotor and energized stator poles. This means that the rotor will move in a clockwise direction until it is lined up with pole faces *b* and *f*. (Again, the rotor will oscillate about *b* and *f* before coming to rest.) If we now energize coils *c* and *g* the rotor will step again in a clockwise direction. If, on the other hand, we energize poles *a* and *e* the rotor will step back in a counterclockwise direction. The motor, in other words, is reversible, depending upon the sequence with which we energize the stator coils.

The motor shown in Figs. 14–10 and 14–11 would require an "eight phase" drive circuit. In other words, we would have to energize the stator coils in the eight-step sequence, shown in Table 14–1, to produce one full clockwise revolution of the rotor.

Note that the sequence starts to repeat after eight steps. Notice too, that each coil is eventually energized in two directions, with first a north, and then a south pole at its inner end. This means that the direction of current flow in each stator pole is important and must be sequenced properly and, therefore, a fairly complex drive circuit would be required.

The permanent magnet rotor provides a detenting flux (as in Fig. 14–10) even when the last drive coil

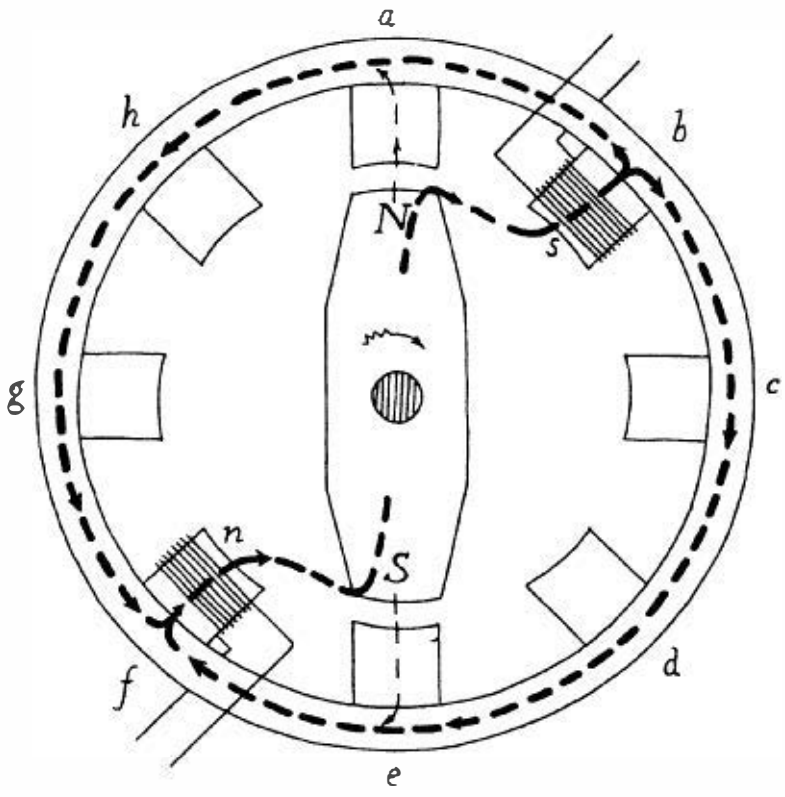


Fig. 14–11. Flux pattern produced in the motor of Fig. 14–10 when coils and *b* and *f* are energized. The rotor will move clockwise to reduce the length of these flux lines.

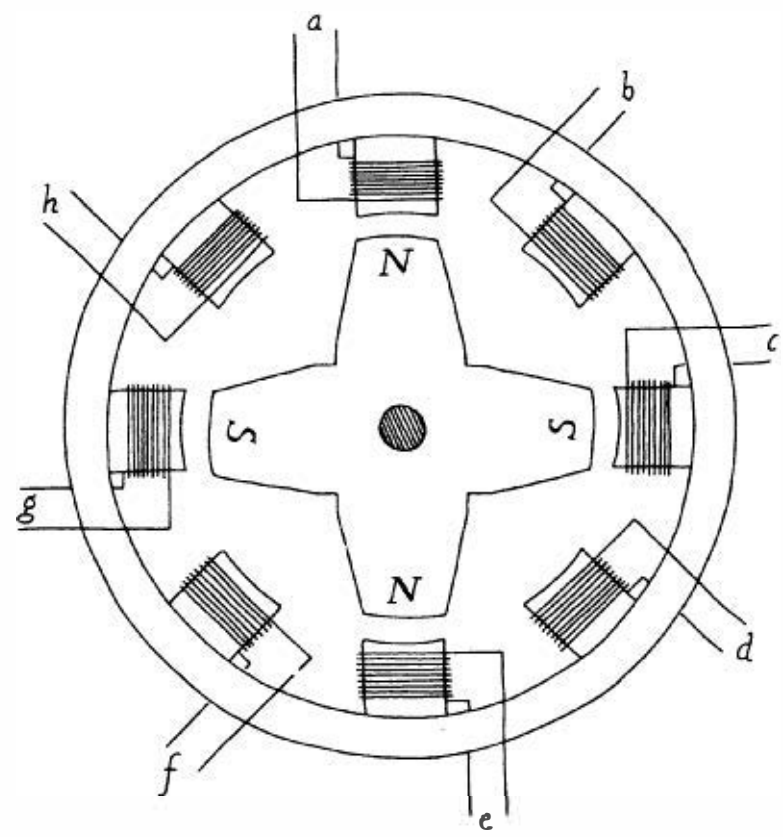


Fig. 14–12. Permanent magnet stepping motor with four poles and requiring only a four-step drive sequence.

Table 14–1. Eight-step Sequence for Energizing Stator Coils in a Stepping Motor

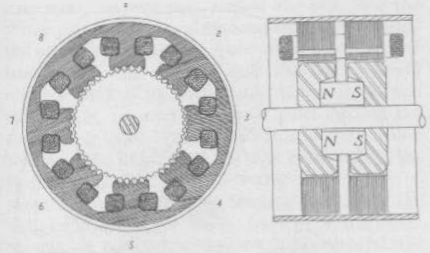
Step Number	Coil							
	a	b	c	d	e	f	g	h
	Polarity							
1		S				N		
2			S				N	
3				S				N
4	N				S			
5		N				S		
6			N				S	
7				N				S
8	S				N			
1		S				N		

is turned off. De-energized detent torque is a characteristic of most permanent-magnet steppers.

In Fig. 14–12 the rotor has been altered so that it has two north poles and two south poles instead of only one of each. The clockwise drive sequence required for this design repeats after four steps instead of eight as shown in Table 14–2.

Table 14–2. Four-step Sequence for Energizing Stator Coils in a Stepping Motor

Step Number	Coil							
	a	b	c	d	e	f	g	h
	Polarity							
1		S		N		S		N
2	N		S		N		S	
3		N		S		N		S
4	S		N		S		N	
1		S		N		S		N



Drawings courtesy of The Superior Electric Company

Fig. 14–13. End view (*Left*), and cross-section drawing (*Right*), of a large stepping motor with permanent magnet rotor.

The four-pole rotor will also produce greater drive torque, of course. By selecting the number of stator and rotor poles the designer can alter such things as torque, stepping angle, complexity of drive circuits, etc.

Permanent magnet stepping motors are generally considered to be rather simple devices, characteristically producing fairly large stepping angles with lower torque and at lower stepping speeds than can be obtained with variable reluctance motors, which will be discussed later. They do not take such precise steps as some other types, although their stepping accuracy is good enough for most applications. In general, they exhibit less tendency to overshoot than do other types of steppers, so that damping is less often required. This is, in part, because the rotor is a permanent magnet and will generate hysteresis effects and eddy currents in unused poles as it turns; this damps the motion. Rotor inertia is typically higher with a permanent magnet design than with a variable reluctance design.

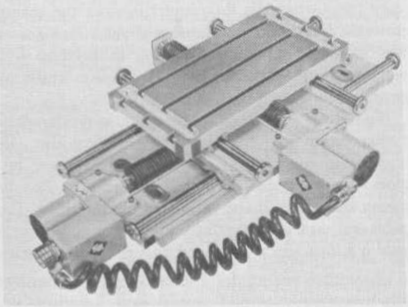
Generalities of this sort, of course, are refuted, eventually, by skillful designers. Figure 14–13 (*Left*) shows the construction of a stepping motor with a permanent magnet rotor that generates high torque,

Photograph courtesy of The Superior Electric Company

Fig. 14–14. Major components of the stepping motor shown in Fig. 14–13.

takes small steps, operates at high speed and has high resolution (good step-angle accuracy). The rotor is magnetized axially rather than radially, as can be seen in Fig. 14–13 (*Right*), the side view of this design. There are two sets of rotor teeth, one set on each end of the rotor. One of these sets is magnetized north, the other south. The two sets are angularly offset by one-half-a-tooth pitch so that as one end of the rotor is seeking north poles, the other end of the rotor can be seeking south poles. As a result, everything is working together and the device produces high torque.

Again in the left illustration, notice that the rotor is aligned with the center tooth on stator poles 1



Photograph courtesy of The Superior Electric Company

Fig. 14–15. *XY* table for machine tool driven by two of the stepping motors shown in Figs. 14–13 and 14–14.

and 5, is slightly misaligned with poles 2, 6, 4 and 8 and is seriously misaligned with 3 and 7. Four switching operations are required to advance the rotor of this design one tooth pitch. With 50 teeth on the rotor, 200 switching operations are required to produce one revolution; but there are only four different switching operations involved, thus the drive circuit is not too complex.

Figure 14–14 shows the major components of this type of motor while Fig. 14–15 shows it in use. Two motors are used to provide the *X* and *Y* drives for a machine tool indexing table.

Variable Reluctance Motors

A different type of stepping motor is shown in Fig. 14–16; it is called a variable reluctance motor. There

are no permanent magnets involved in the design shown (although permanent magnets can be used to produce some of the stator fields in certain variable reluctance motor designs). The rotor is a piece of soft iron. It has been positioned as is shown in Fig. 14–16, by energizing coil *a*. Energizing coil *b*, as shown in Fig. 14–17, will produce the flux pattern shown by the heavy and light dotted lines. (It makes no difference which direction the current takes in the drive coils.) Notice that there is a heavy flux produced in the gap between stator *a* and the rotor, but that there is no bending of the flux at this point; no rubber band to straighten out. Therefore, there will not be any torque developed between stator *a* and the rotor. There is no pole overlap between stator pole *c* and the rotor; and the air gap here is very large, thus the flux, and therefore the torque, is weak. Furthermore, the flux divides and goes in two different directions, tending to produce both clockwise and counterclockwise torques which approximately cancel each other.

Significant drive torque, however, is produced between stator pole *b* and the rotor. Here the poles overlap in such a way as to drive the rotor in a clockwise direction. As motion occurs, the flux between stator *a* and the rotor will become bent, (the poles will start to overlap) producing a torque at pole *a* which tends to retard this clockwise motion.

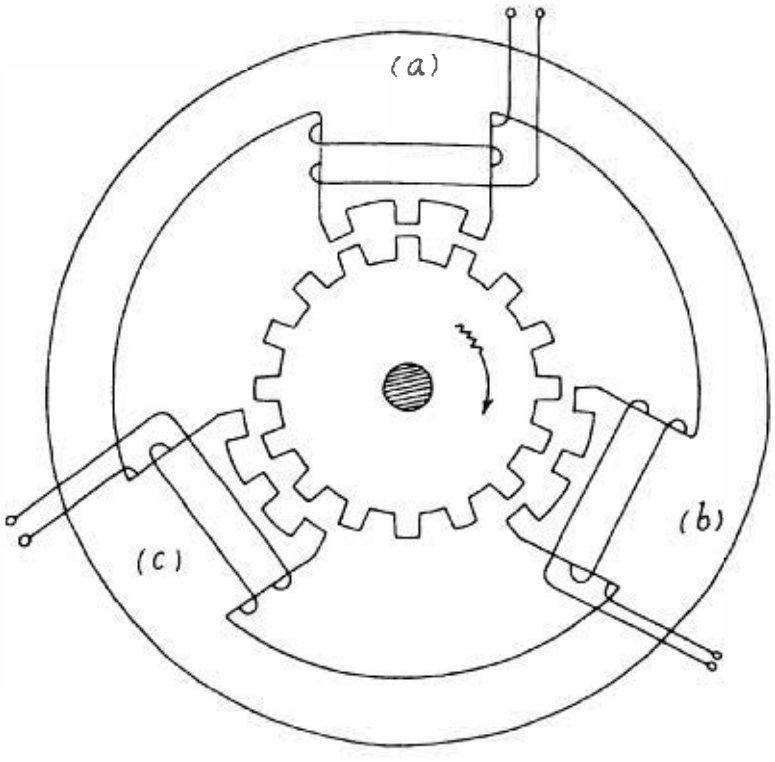


Fig. 14–16. Schematic illustration of a typical, variable reluctance stepping motor with rotor of soft iron and no permanent magnets in the design.

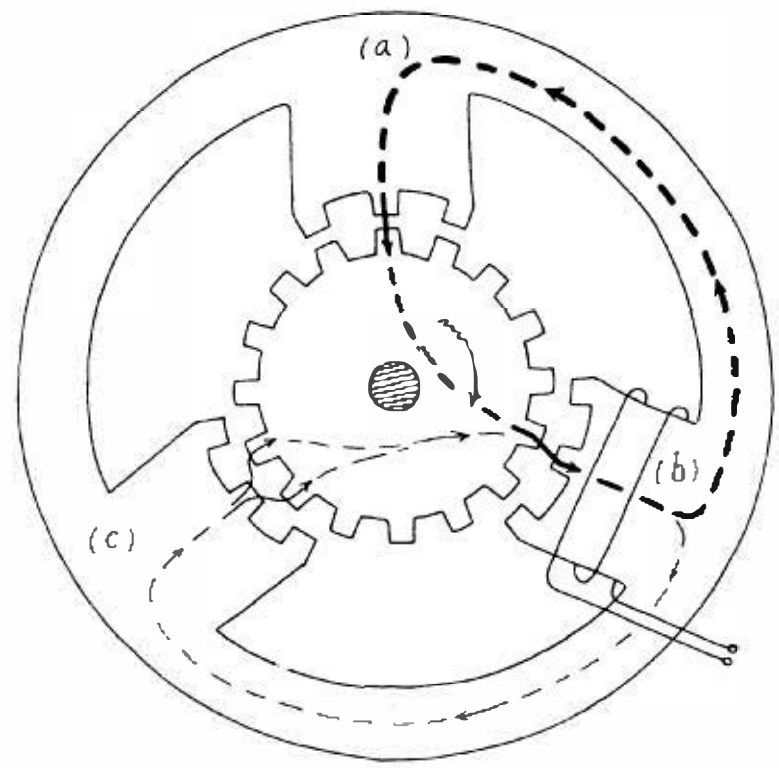


Fig. 14–17. Flux patterns produced by energizing stator pole *b*, in the design shown in Fig. 14–16.

But there is more flux at stator *b* than at either stator *a* or *c* (the combined *a* and *c* flux passes through *b*), thus *b* will dominate. As a result, the rotor will finally align itself on *b*.

Variable reluctance motors typically have small stepping angles, operate at high speeds, produce high torques, and frequently require damping. The design shown in Figs. 14–16 and 14–17 would have no de-energized detent torque, since no permanent magnets are available to provide such torque. Coil *b*, of course, could be kept energized until we were ready to energize coil *c*, and produce the next step. An "energized detent" of this type is sometimes called *holding torque* rather than *detenting torque*, since it consumes power. As pointed out earlier, however, such a detent can be turned off when it is desired to take the next step.

Electro-Hydraulic Pulse Motor

Figure 14–18 shows another configuration for a variable reluctance motor. The rotor is a disc with alternately thick and thin pie-shaped teeth. The stator poles consist of two rings also with alternately thick and thin pie-shaped teeth. The rotor-stator pole assembly is surrounded by a single electrical coil, as shown in the illustration. This coil will produce a toroidal, or doughnut shaped flux field (as shown by the dotted lines) when it is energized.

The flux lines will pass around the coil, into the stator pole ring on the right-hand side, then through the rotor disc,into the left-hand stator pole ring, and back around the electrical coil. The flux lines will seek the path of least reluctance as they pass from stator pole, to rotor, to stator pole; seeking the smallest air gaps. When the rotor is in the position shown in the illustration, the flux produced by the electrical coil will tend to rotate the rotor in a counterclockwise direction (looking at it from the right-hand end of the shaft). Once rotor and stator are aligned, however, subsequent applications of current to the electrical coil will not produce any further motion, thus this system is capable of taking only one step; and then only if it is misaligned as shown, to begin with. (See also Figs. 14–19 and 14–20.)

If five such rotors, each misaligned from the others but all fastened to the same shaft, are assembled in a single housing, however, and if the stator coils are energized in the proper sequence, a very good stepping motor can be constructed. Such an arrangement is shown in Fig. 14–19. This is a very powerful high-speed device producing one-half horsepower at speeds of up to 16,000 pulses per second. That's right, per second, not per minute!

The designers of this motor, however, have gone still further and have provided an hydraulic torque amplifier for applications requiring these extreme stepping speeds and higher output power (up to 10 horsepower). This system is shown in Fig. 14–20. The variable reluctance stepping motor is shown on

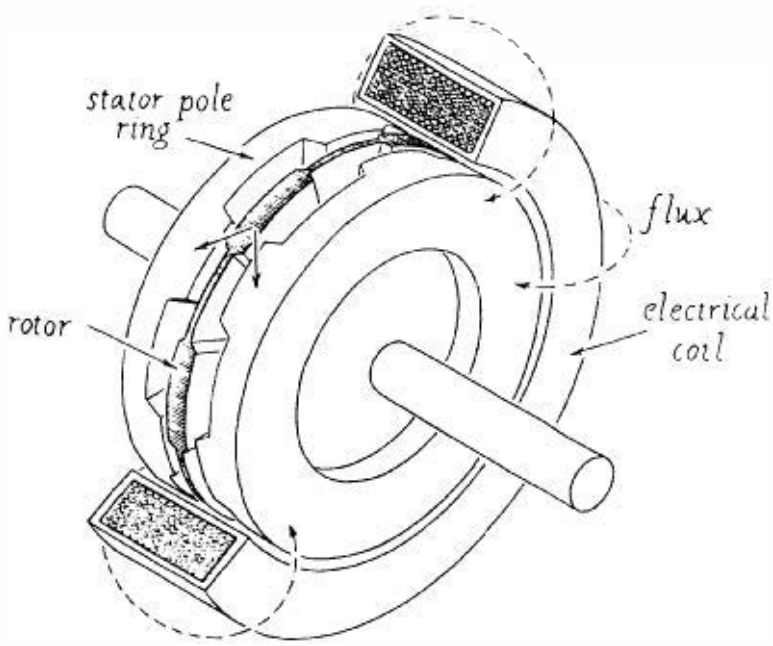
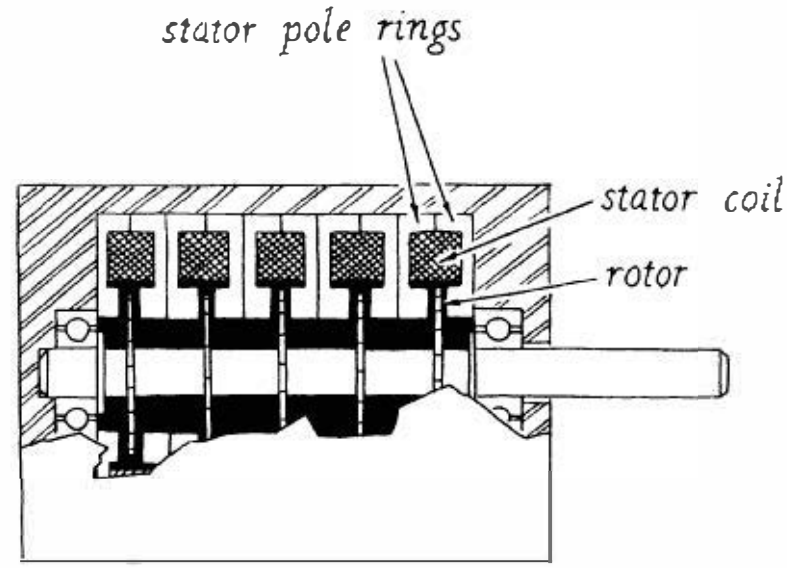


Fig. 14–18. Partial view of a type of variable reluctance motor. The section shown above would only be capable of taking a single step, and then only if initially misaligned as shown. (See also Figs. 14–19 and 14–20).



Drawing courtesy of the ICON Corporation

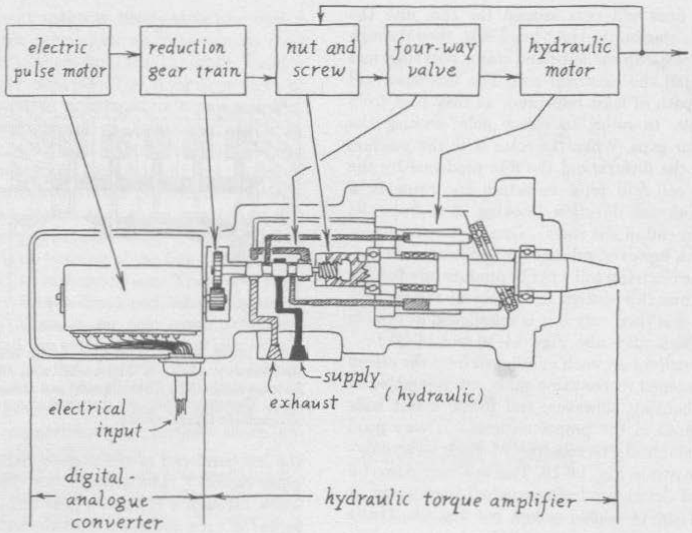
Fig. 14–19. Electrical stepping motor comprised of five of the "motors" shown in Fig. 14–18, with all rotors displaced from each other (by three degrees) and mounted on a common shaft. The stator coils are energized in sequence to produce intermittent rotary motion.

the left-hand end of the system (labeled "electrical pulse motor"). The pulse motor drives a nut and screw through a reduction gear train. The nut and screw, in turn, control a four-way valve which feeds a powerful hydraulic motor, the valve and hydraulic motor combination serving as an hydraulic torque amplifier to drive the ultimate load. An hydraulic feedback loop is provided to guarantee that the hydraulic motor accurately obeys the commands of the electrical pulse motor. The performance of this electrohydraulic stepping system is very impressive. Figure 14–21 is a photograph of one of the several models available.

Flexible Spline Motor

Figure 14–22 is a schematic representation of a very unusual stepping motor. A flexible spline is located inside a rigid ring gear. Certain portions of the spline are pulled radially outward by stator coils, to engage the ring gear. The spline has a different pitch than the stator, thus the teeth on the spline are slightly misaligned with the stator teeth, and therefore, as the spline moves outward, it rotates slightly as it aligns itself with the stator. Spline and stator teeth only engage near the energized coils as can be seen from the illustration.

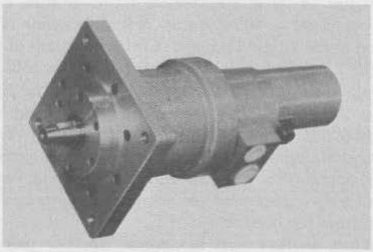
When another step is desired, another pair of stator coils is energized and a different portion of the spline engages a different portion of the ring gear. Again there is initial misalignment between



Drawing courtesy of the ICON Corporation

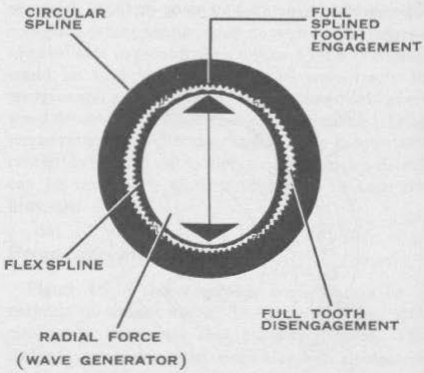
Fig. 14–20. Electro-hydraulic pulse motor. In this device the output of the stepping motor of Fig. 14–19 is amplified by an hydraulic torque amplifier with the resulting system producing up to 10 horsepower at 16,000 steps per second.

spline teeth and stator teeth (since spline and stator have different pitches), causing additional rotation of the spline. In this way, the spline is flexed and forced to rotate by sequential energization of the stator coils. The action is really very similar to that of the inverse escapement shown in Fig. 12–33, even



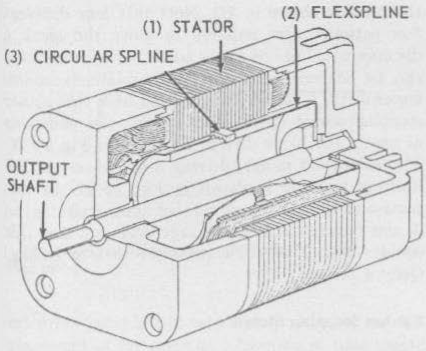
Photograph courtesy of the ICON Corporation

Fig. 14–21. Photograph of the electro-hydraulic stepping motor shown in Fig. 14–20.



Drawing courtesy of the United Shoe Machinery Corporation

Fig. 14–22. Schematic illustration of a flexible spline stepping motor. The principle of operation is very similar to that of the inverse escapement shown in Fig. 12–33.



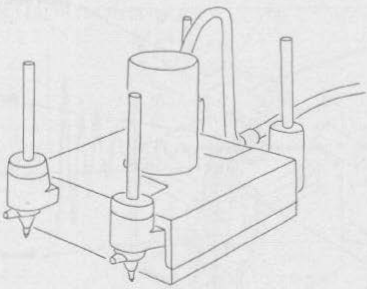
Drawing courtesy of the USM Corporation

Fig. 14-23. Cutaway drawing of the flexible spline stepping motor shown in Fig. 14-22.

though the two systems differ from each other in all important details. Figure 14-23 shows a cutaway drawing of the flexible spline stepper. Note that unlike more conventional steppers this one does involve metal-to-metal contact between stator and rotor.

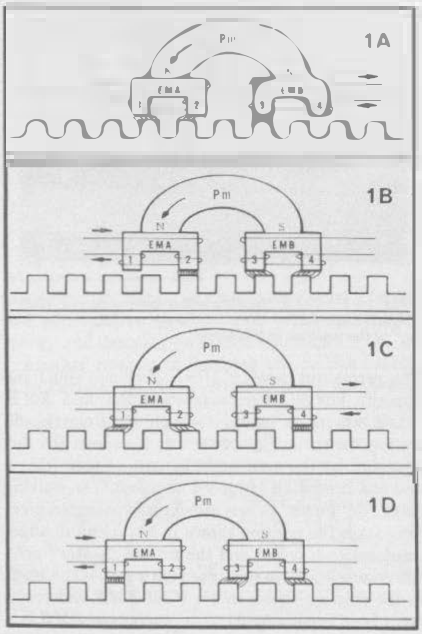
Planar Stepping Motor

Another unusual stepping motor (patented both in the U.S. and abroad) is shown in Fig. 14-24. This is a permanent magnet stepper in which the "stator" is a large table and the "rotor" is a small block that will move in either an X or Y direction (or both simultaneously) over the surface of the table. Small electromagnets are used to switch the flux from a permanent magnet, for each axis in the block, to produce the stepping motion. The surface of the table over which the block moves consists of rows and columns of square teeth as shown in Fig. 14-25.



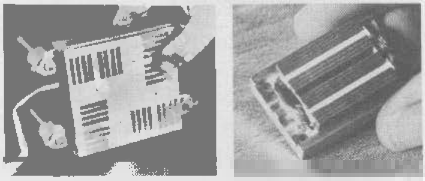
Drawing courtesy of Xynetics, Inc.

Fig. 14-24. Planar stepping motor that can be indexed in either the X or Y direction (or both simultaneously) over a "stator" table top covered with poles.



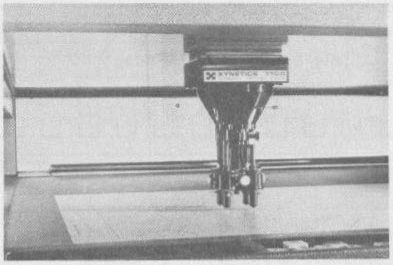
Drawing courtesy of Xynetics, Inc.

Fig. 14-25. Further details of the principle of operation of the planar stepping motor. 1A, B, C, and D—Single-axis, two-phase forcer operation.



Photograph courtesy of Xynetics, Inc.

Fig. 1[illegible] (*Left*) A [illegible] drive head from a planar stepping motor. (*Right*) Bottom [illegible] single [illegible] head.



Photograph courtesy of Xynetics, Inc.

Fig. 14–27. Planar stepping motor used in an *XY* plotter. Magnetic forces allow the planar motor to cling to the "ceiling" of the machine in this model.

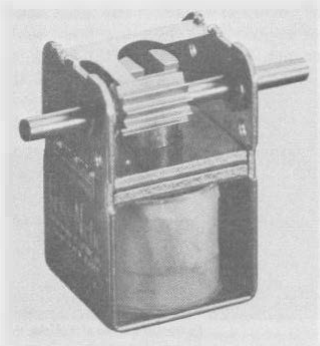
The permanent magnet (*Pm*), produces equal and opposite flux in electromagnets *EMA* and *EMB*. When current is passed through *EMB* in the direction shown in Fig. 14–25–1A, it cancels the flux produced by the permanent magnet at pole (tip) 4 and reinforces that produced at pole 3. The walking block (or "forcer" as it is called by its manufacturer) then takes the position shown in 1A, with full alignment between pole 3 and the nearest "stator" pole. Energizing *EMA* to reinforce pole 2 moves the block to the position shown in 1B (Coil *EMB* had previously been de-energized). Next energizing *EMB* in a direction to reinforce pole 4 and de-energize pole 3 moves the block to the position shown in 1C; while [illegible] *EMA* in the reverse direction results in the position shown in 1D. [illegible]te that four different flux patterns were required to move the block a [illegible]e equal to one stator tooth, a high resoluti[illegible] can be [illegible], therefore, with relatively coarse stator teeth. A complete drive head from [illegible] planar stepping motor (*Left*), and (*Right*) the underside of one of th[illegible] shown in Fig 14–26 The completed motor, driving a stylus on an *XY* draft[illegible] shown in Fig 14–27. In this particul[illegible] the motor n[illegible] only walks in an *X* and [illegible] direction over the large table, but does hi[illegible] upside down. The table in this case is a ceiling! Quite a gadget

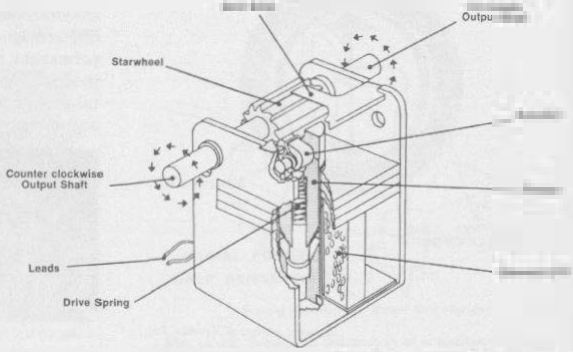
[illegible]

Figure 14–28 [illegible]s another type of [illegible]g motor, [illegible] one [illegible]n ratchet motor A cutaway [illegible] of th[illegible] pa[illegible] design was seen in Fig. 7–34, in the chapter on ratchet mechanisms Of small size and offering very high torque at low cost, it has [illegible]s place, but its advantages and disadvantages, operat[illegible]g features, etc , are those of a ratchet motor subject to impact [illegible]ion, mechani[illegible]or, etc It cannot therefore be considered a [illegible] stepping motor and will not be further discussed here.

[illegible]

When a permanent magnet, variable reluctance, or electro-hydraulic steppi[illegible] motor is fed an elect[illegible]





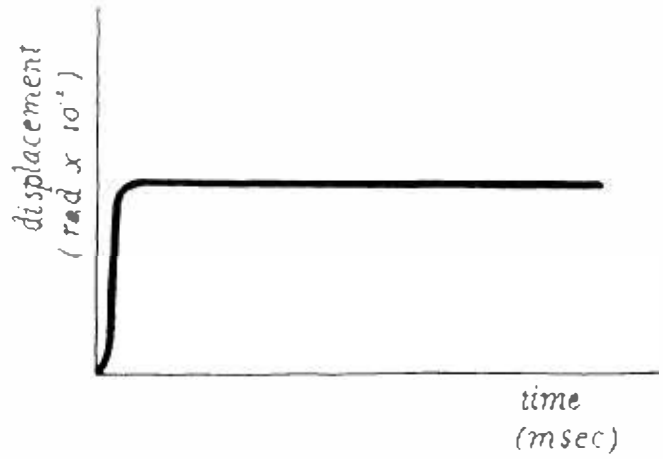


Fig. 14-29. Ideal single-step displacement curve for a stepping motor.

cal drive pulse it will take a step. Ideally, the displacement of the rotor as a function of time should be that shown in Fig. 14–29. Since all contact between rotor and stator consists of magnetic flux, (which, as has been seen, acts very much like a large elastic band), the displacement-versus-time curve is more apt to look like that shown in Fig. 14-30. The electrical pulse "impulses" the rotor which attempts to take a smooth step, but which responds (as did the impulsed block-on-a-spring in Fig. 3–19) by overshooting, then undershooting, then overshooting, then undershooting, etc., before finally reaching the new position. The response is that of a damped vibrating system to an impulse or shock input. If the damping on the output is increased, the rotor will settle more rapidly, as shown in Fig. 14–31. Various types of dampers have been proposed for this application, including mechanical devices such as that shown in Fig. 14-32, where a large inertia disc (*20*) is loosely coupled to the output shaft (*14*) of the stepping motor; the interconnection

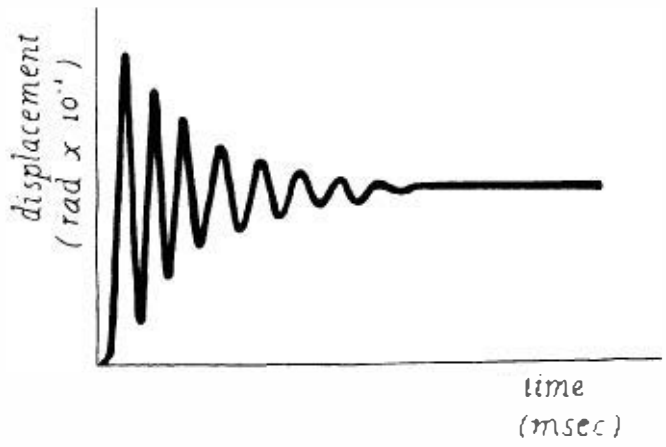


Fig. 14-30. Typical step for a conventional stepping motor. Because the only connection between rotor and stator is in the form of magnetic flux, the system responds like a vibrating spring-mass to single impulse (shock) excitation. (See also Fig. 3-19.)

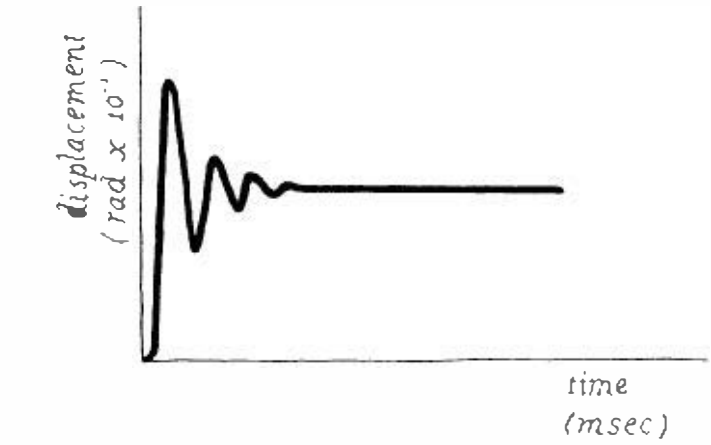


Fig. 14-31. Response of a highly damped stepping motor.

being through friction or viscous drag only. The disc will slip relative to the motor shaft, dissipating energy and damping unwanted vibrations.

Another method of damping the motion rapidly is to energize all stator drive coils simultaneously.

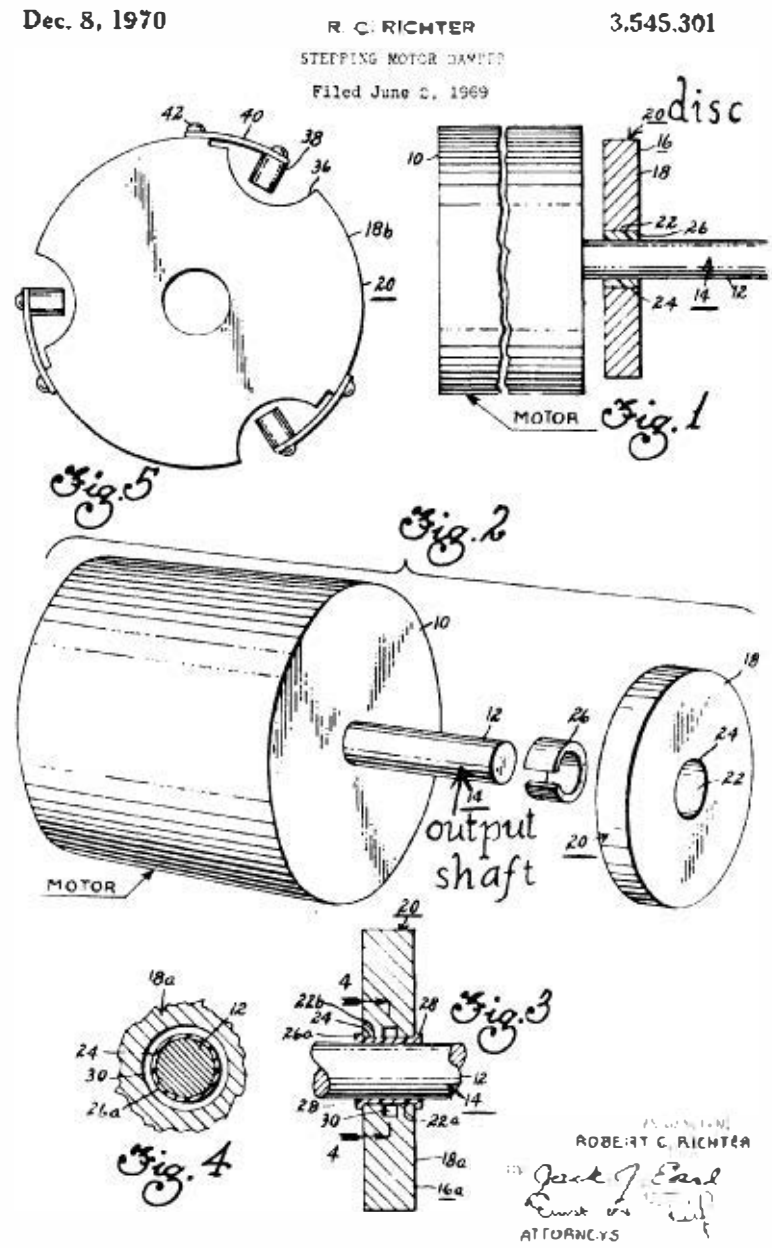


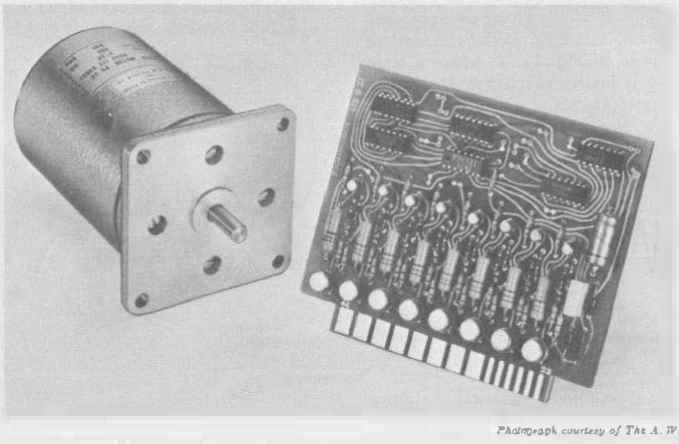
Fig. 14-32. Inertia-friction damper for stepping motor. (U.S. Patent 3,545,301; R. C. Richter.)

Photograph courtesy of The Superior Electric Company

Fig. 14–33. [illegible] stepping [illegible] One of the features of this drive unit is that it will stop the motor after a predetermined number of steps have been taken.

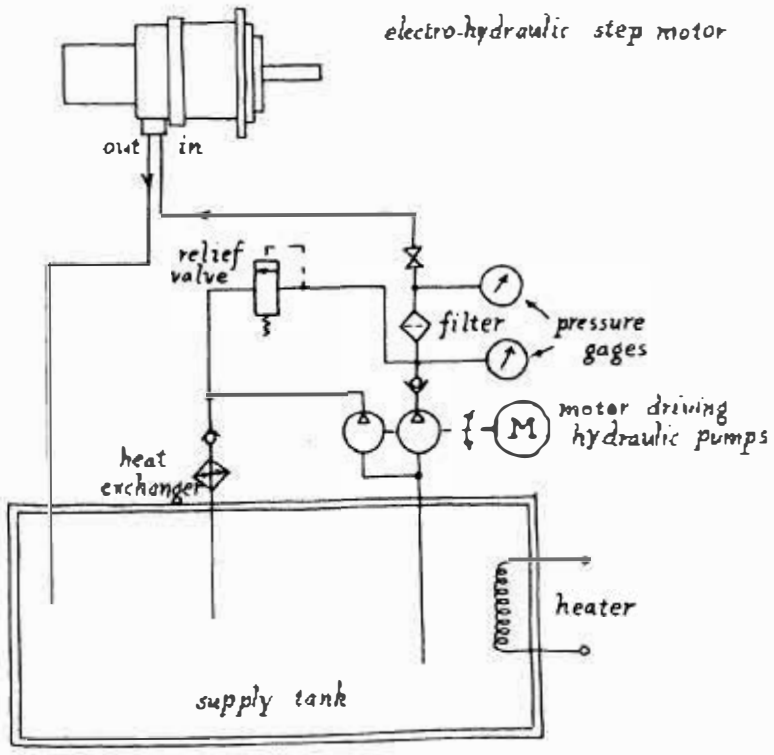
The rotor does not know which way to go and stops quite [illegible] the "unused" stator coils are energized only partially to produce a drag on the rotor rather than employing a complete brake action. At other times, a [illegible] pulse is applied to the [illegible] winding just before the rotor reaches the present winding. In any event, the motion curves of the "typical" stepping motor are free from shock and impact. Accelerations and decelerations are gentle compared to those found [illegible] drives. As a result, stepping motors are frequently encountered in those intermittent motion applications requiring high torque and [illegible] stepping rates.

The flexible-spline-and-ratchet motors, of course, cannot "hunt," thus their motion curves are different than those of the more [illegible] motors



Photograph courtesy of The A. W. Haydon Company

Fig. 14–34. Packaged electronic circuit that will convert a single train of pulses into the appropriate coil energization sequence (or [illegible] separate, properly [illegible] pulse trains) to drive a stepping motor.



Drawing courtesy of the ICON Corporation

Fig 14-35. Partial hydraulic circuit required for an electro-hydraulic stepping motor. An electrical control circuit for the electric pulse motor is also required, but is not shown here.

Drive Circuits

The design of electrical circuits is certainly beyond the scope of this book, but the intermittent motion designer should be aware that he can use many different types of drive circuits with most stepping motors. He should consult motor manufacturers for specific recommendations. Many manufacturers, in fact, sell packaged drive circuits such as those shown in Figs. 14-33 and 14-34. Some of these can be preset to stop the motor after a certain number of steps. Some start and stop the motor with relatively slow stepping rates, operating at maximum stepping rate only after the motor and load have been brought up to speed. Some motor drive circuits are intended to produce special control or damping effects, such as those described earlier. Some are designed to speed up stepping rates by "electrical forcing of the drive coils." (See Chapter 13 for further discussion of electrical forcing.) Some circuits are designed to operate a motor at minimum energy levels, others at maximum performance levels which are often produced by considerably increasing energy consumption.

Electro-hydraulic stepping motor systems require more than electrical control circuits; they also require hydraulic circuits for the torque amplifiers. Figure 14-35 shows a typical hydraulic circuit. This emphasizes, I think, that the stepping motor, like a clutch-brake combination, is a system component. Taken alone it cannot produce intermittent motion. With the proper supporting equipment, however, it can produce intermittent motion with a speed-torque capability, and with a control flexibility, unthinkable for the simpler mechanisms discussed in earlier chapters.

Anatomy of a Stepping Motor Indexing System

Following is a brief description of the components and functions of the typical stepping motor indexing system, as shown in Fig. 14-36:

A. Stepping motor; B. Friction or viscous and/or inertial damper to reduce time required to suppress motor-load oscillations between steps; C. Load; D. External controls—can be almost anything from a simple start-stop switch to a digital computer. Provides basic commands: Index, Dwell, Take 276 steps, 180 degrees, etc.; E. Digital logic circuits—probably solid state electronics but could be relays, stepping switches, etc. Converts basic commands into detailed, sequenced electronic circuit commands —for example: Start variable rate oscillator, generate 20 pulses, gradually increasing oscillator rate to 1000 pulses per second, hold at 1000 pps for 674 steps, gradually . . . etc.; F. Programmer—If stepping distance or sequence must be changed frequently it may be desirable to have a programmer to aid the logic circuits. This would probably utilize solid state electronics, but could use groups of switches, punched cards, tape or equivalent. If the program is simple and constant, it can be "hard-

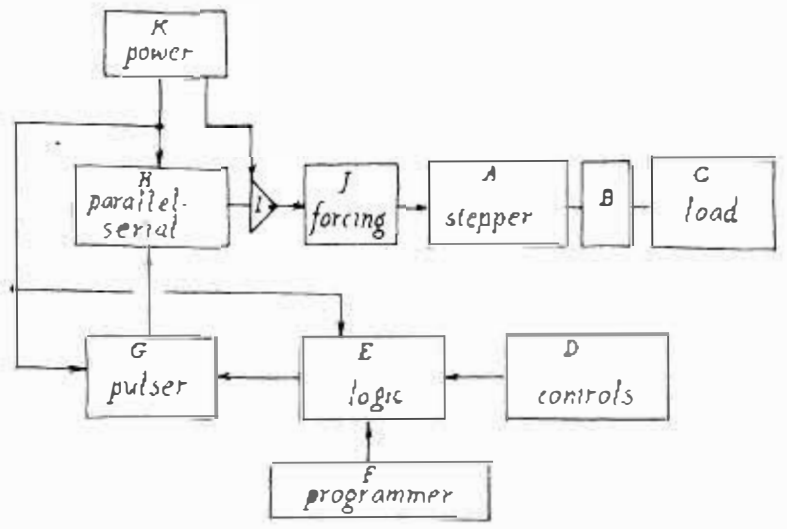
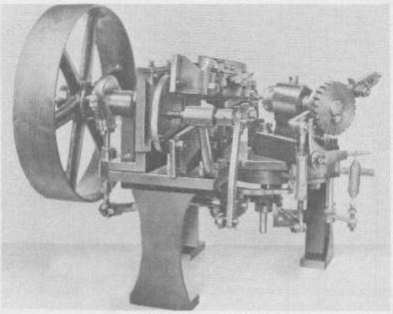


Fig. 14-36. Anatomy of a stepping motor indexing system.

wired" [illegible] the logic [illegible]s; G [illegible] rate pulse generator—Logic level electrical pulses at a controllable rate to allow gradual acceleration and [illegible] of heavy loads. As an alternate this can be an [illegible] magnetic or optical pulser driven by a machine in some other part of the total system; H Serial to parallel converter—Receives [illegible] pulse train from *G* and [illegible] into [illegible] or 8 pulse trains for the stepping motor; I [illegible]fier—Receives logic level pulse trains from *H* and amplifies them to the power level required to drive the motor; J. Forcing circuits—Series resistors parallel capacitors, etc., for reducing the electrical time constant of the [illegible] stepping motor to increase stepping rates; K DC power supply [illegible] logic level power for F G and H and power levels for I.

[illegible]

Stepping motors are widely used today in machine tools and production machinery where high-speed precision indexing of heavy loads is a frequent requirement. They are, therefore, one of the latest types of intermittent motion devices to be used for these applications. The first mechanisms used were probably ratchets with hand-index, or manually clutched and braked indexing tables or both. Figure 14–37 shows the original Robey-Smith bevel gear planer, built in 1895. The gear being cut is positioned by a bidirectional ratchet similar to that shown in Fig. 7–16.



British Crown copyright, Science Museum, London

Fig. 14–37. Robey-Smith bevel gear planar built in 1895.

CHAPTER 15

Miscellaneous Types

All of the major classifications of intermittent motion mechanisms and systems have now been considered: gears, ratchets, cams, Genevas, stepping motors, escapements, etc., but the subject is not yet complete. Since intermittent motion devices are used in many different ways, special or unusual features are often required. The various common solutions will not solve all of the problems nor will they fit all possible applications. Designers have, therefore, evolved many other devices and systems for producing intermittent motion. Some of these are ancient and yet have never found mass application. Others are more recent and may some day become as popular as Genevas, ratchets, and the more common mechanisms. In this chapter we will consider a few of these, as yet, uncommon devices.

Star Wheels

Figure 15–1 shows two common intermittent motion mechanisms we have considered previously: a pair of gears, one of which has been mutilated, and a five-slot Geneva. Both are simple and popular devices and each has advantages and disadvantages. The Geneva mechanism, for example, has a relatively attractive acceleration and deceleration pattern; at least when compared to mutilated gears which produce severe impact. Mutilated gears, on the other hand, are capable of producing a long motion cycle interrupted by a relatively short dwell period. They can also produce one, two, or three dwells per revolution of the output. The Geneva, of course, produces a fairly short motion period followed by a long dwell period and can usually produce no fewer than four dwell periods per revolution of the output. (Genevas with only three dwells have been made, but are awkward; while one- and two-dwell Genevas are impossible.)

Good ideas always appear to be simple and inevitable once some clever designer has thought of them. In this next case, someone decided to combine the better features of a Geneva and a mutilated gear. The result is called a star wheel, and an early example is shown in Fig. 15–2. A drive member similar to a Geneva driver, but containing five drive pins instead of one, indexes an output wheel. This wheel has "starting slots" that are similar to the drive slots of a Geneva wheel, although they are curved rather than straight. The first pin on the driver engages one of these slots and starts the output wheel in motion. The other pins on the driver then engage the gear teeth and continue the drive motion until the output member is brought to rest by Genevalike locking surfaces on input and output members.

In the earliest star wheels the starting slots were epicyclic curves. Figure 15–3 shows a modern star wheel in which the slots are circular arcs and involute gear teeth have replaced the drive pins for the constant velocity portion of the motion cycle. Using circular arcs instead of epicyclic curves makes the star wheel a lot easier to manufacture. The use of involute teeth instead of pins also helps and it is felt that these improvements make the star wheel a practical device for the first time.

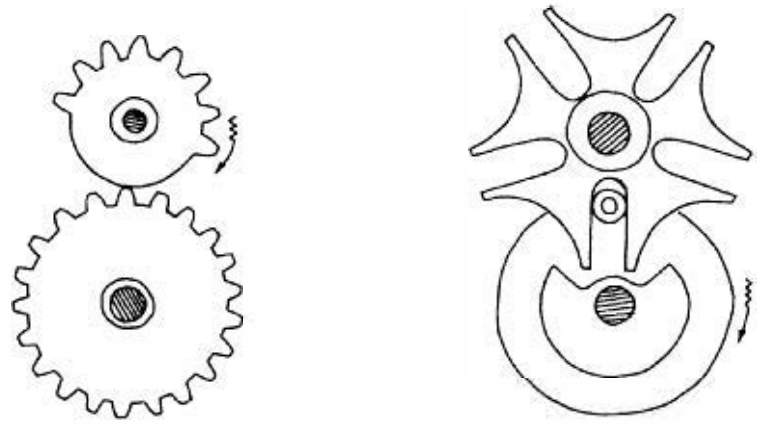
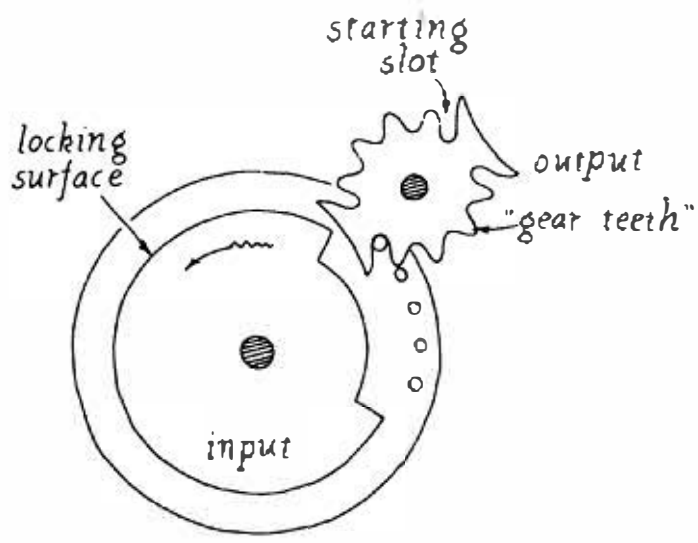


Fig. 15-1. Two common intermittent motion mechanisms; a mutilated gear and a four-slot external Geneva. The star wheel combines some of the good and bad features of both types.

Other versions of the external star wheel, producing between one and six stops per revolution of the output member can be seen in Fig. 15–4. Note that gear segments are not included on the star wheels with more than two stops per revolution, but that the curved "starting slots" are still present. A star wheel with four stops, therefore, is more comparable to a Geneva than are the one- and two-stop versions. The advantage of the four-slot wheel over a Geneva lies in the acceleration pattern. The dwell motion ratio of the input of these star wheels can also be varied somewhat, as with a Geneva.

Acceleration curves for one quarter cycle of revolution of a four-slot external Geneva and a four-stop star wheel are compared in Fig. 15–5. The star wheel has a smaller peak acceleration than the Geneva, but has a larger instantaneous change in acceleration at the beginning and end of motion.



Drawing courtesy MACHINE DESIGN Magazine; Dec 23, 1965; p. 121 ff

Fig. 15–2. Early star wheel. This is called a two-stop wheel since it will dwell twice for each output revolution.

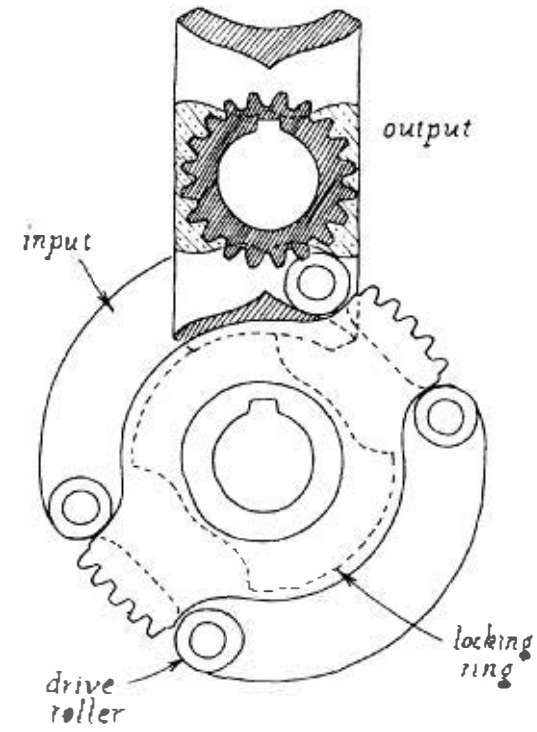
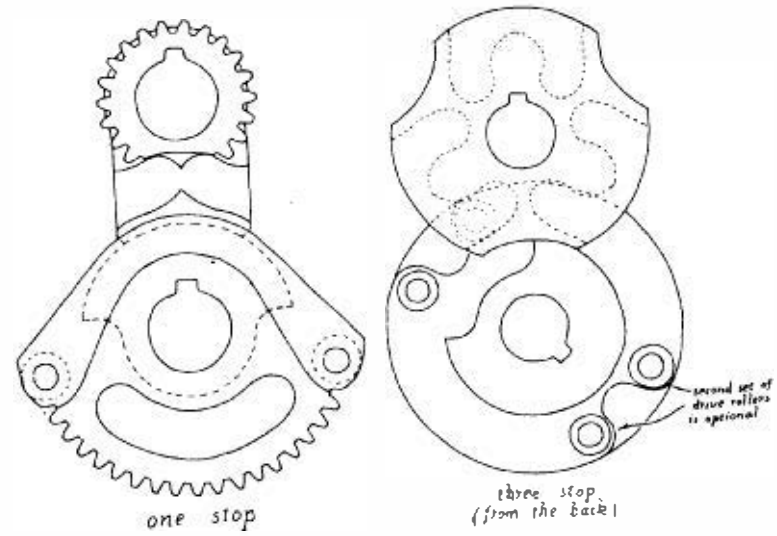
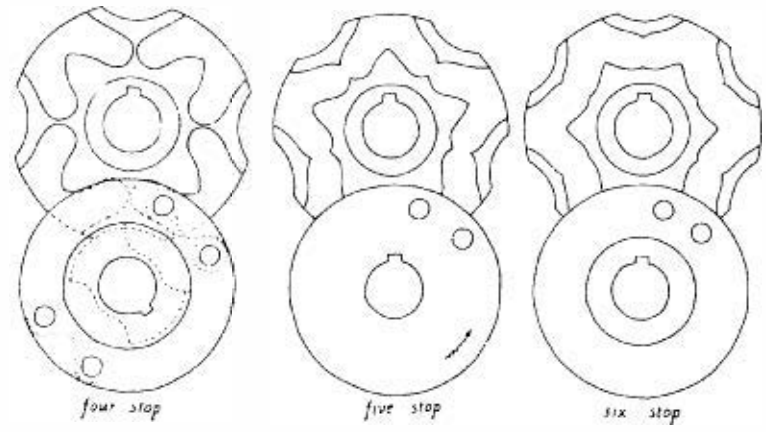


Fig. 15–3. Contemporary two-stop star wheel. (U.S. Patent 3,443,455; M. J. Zugel.)





Drawings courtesy of the Cyclo Index Corporation

Fig. 15–4. One- to six-step external star wheels. Note that the dwell-motion ratio of the input can also be varied somewhat by providing additional drive rollers as with a Geneva.

Theoretically, any instantaneous change in acceleration will produce infinite jerk. In practice, however, a greater instantaneous *change* (or a more rapid rate-of-change) in acceleration will produce more jerk; this is considered a disadvantage of the star wheel by some machine tool designers. The lower maximum acceleration, however, attracts other designers, since torque is proportional to acceleration, not to jerk; and a lower peak acceleration means lower maximum drive torque. A given diameter drive pin, therefore, will see less stress and/or wear less rapidly in a star wheel system than in a Geneva system.

The acceleration curve of a star wheel is altered significantly by changes in the radius of curvature of the starting slots; the curve in Fig. 15–5 being an optimum curve obtained with the optimum slot shape. As with a Geneva, the acceleration curve is also dependent upon the number of dwells per revolution of the output member, with the peak acceleration tending to decrease as the number of dwells is increased (again, similar to a Geneva).

In Fig. 15–6 several types of internal star wheels are shown. In each case the output member rotates in the same direction as the input (counterclockwise), with the lead roller starting the action (accelerating the output) and the second roller stopping it. As indicated in the first part of Fig. 15–6A, 240 degrees of input motion produces 360 degrees of motion in the output. The output then dwells during the next 120 degrees of motion of the input. In the other design shown, 180 degrees input motion produces 360 degrees of output motion. During the next 180 degrees of input motion there is 180 degrees of output dwell. The action is reversable. The other designs, shown in part *B* of the illustration, produce different amounts of output motion for varying inputs. Gear segments cannot be included in an internal star wheel (eliminating one of its advantages over the Geneva), but one, two, and three dwells per revolution of the output are still possible, while with an internal Geneva, this is either difficult or impossible.

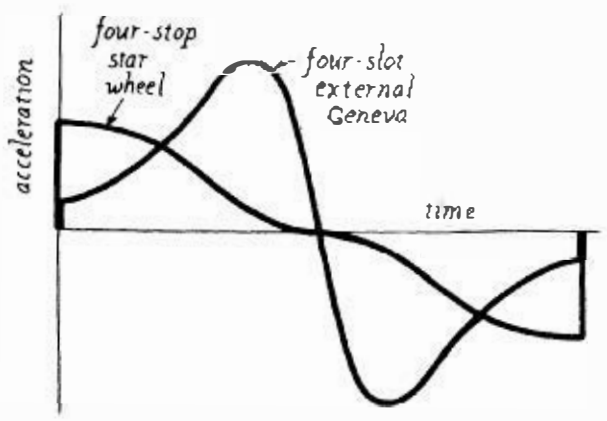
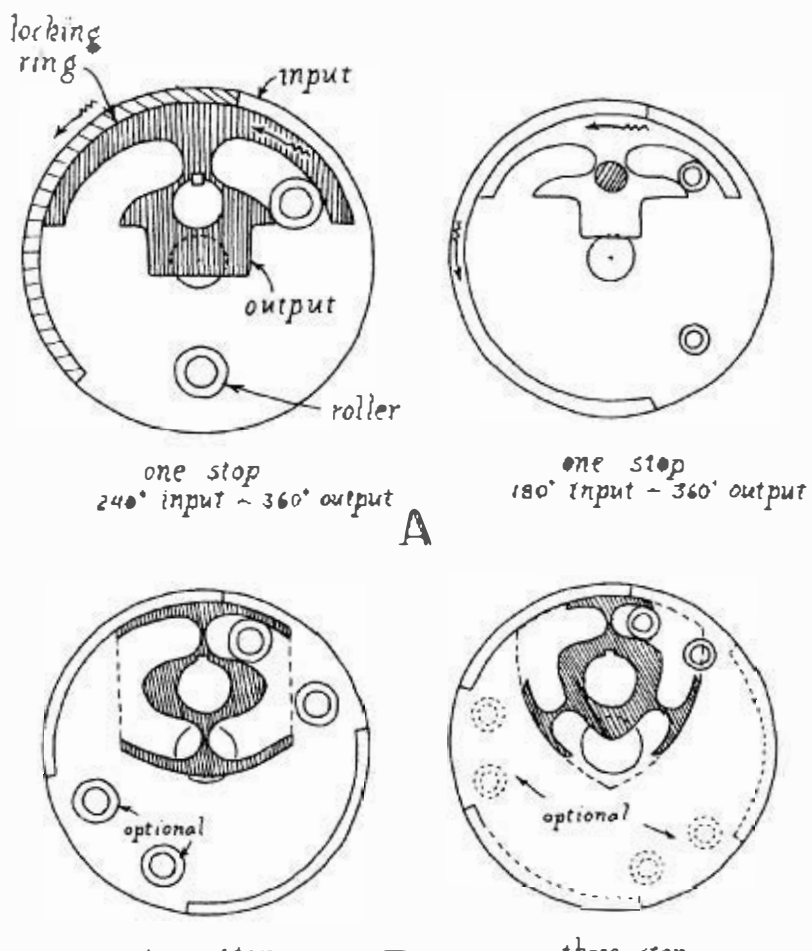


Fig. 15–5. Acceleration curves for a four-stop external Geneva and a four-stop star wheel. Note the initial and peak accelerations produced by each mechanism.



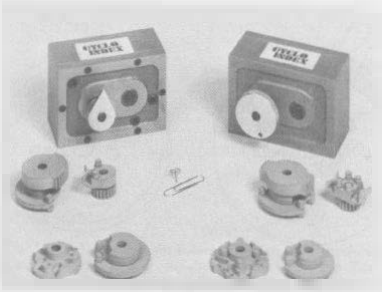
Drawings courtesy of the Cyclo Index Corporation

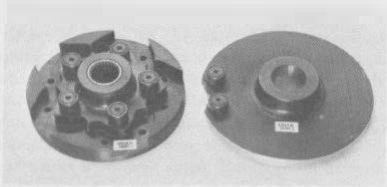
Fig. 15–6. One-, two-, and three-stop internal star wheels.

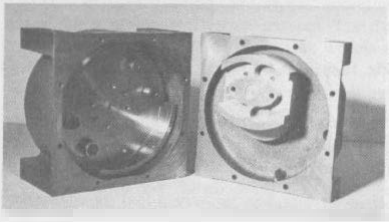
The units of commercially available star wheels shown in the upper photograph in Fig. 15–7, are miniature devices with one-inch center distances between input and output members. The large, five-stop external start wheel (Left, below) is twelve inches in diameter (9½-inch center distance), while the two-stop internal unit (shown at right) has produced over 100 million trouble free (and maintenance free) operations in one application, and is still going strong.

Roll Cam Drives

Electrical typewriters, computer printers and other business machines and peripheral equipment operate at very high indexing speeds—pushing ratchets, escapements, Genevas, and other simple indexing







Photographs courtesy of the Cycle Index Corporation

Fig. [illegible]–7. Con[illegible] star wheels [illegible]e-inch to 9 ½-inch center distances.

devices to their limits. Designers of this type of equipment, therefore, have looked [illegible]rnate systems which would produce high indexing rates with little or no impact. Some of the approaches taken were se[illegible] in Chapter 10, on G[illegible]ng. Another approach, shown schematically in Fig. 15–8, is called a roll cam drive system; a very popular system in contemporary business machinery. The input consists of a high-speed, continuously rotating drum which has a friction surface (rubber or elastomeric coating). The output members do no[illegible] normally engage this roller. When an output [illegible] desired, an electrical signal energizes the small co[illegible]rol solenoid. This actuates a trigger which, in tu[illegible]

eccentric cam. A [illegible]s, on the cam rotat[illegible]s i[illegible] clockw[illegible] the cam engages the input roll. The roll now rotates the cam further. [illegible] rotati[illegible] of t[illegible] cam [illegible] transmitted through a [illegible] of gears to the output shaft [illegible] also acts as a fulcrum for the lever *ln* on [illegible] the friction roller [illegible] [illegible] [illegible]

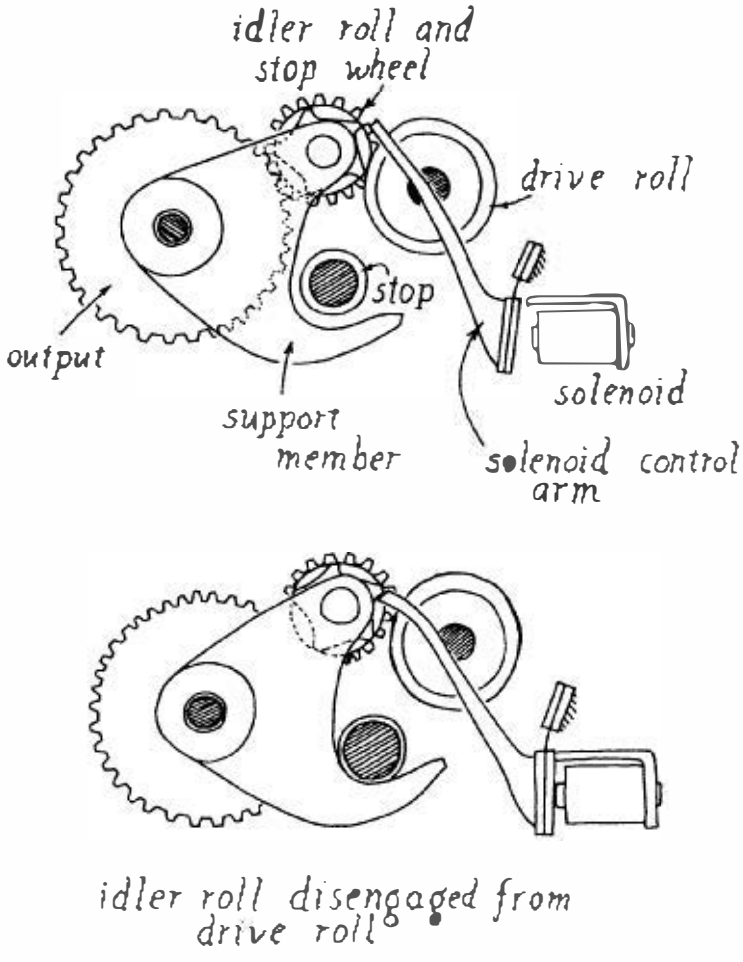


Fig. 15-9. Another type of roll cam driveisystem in which the output can be stopped in any one of ten different positions (U.S. Patent 2,759,672; C. S. Simonds, et al.)

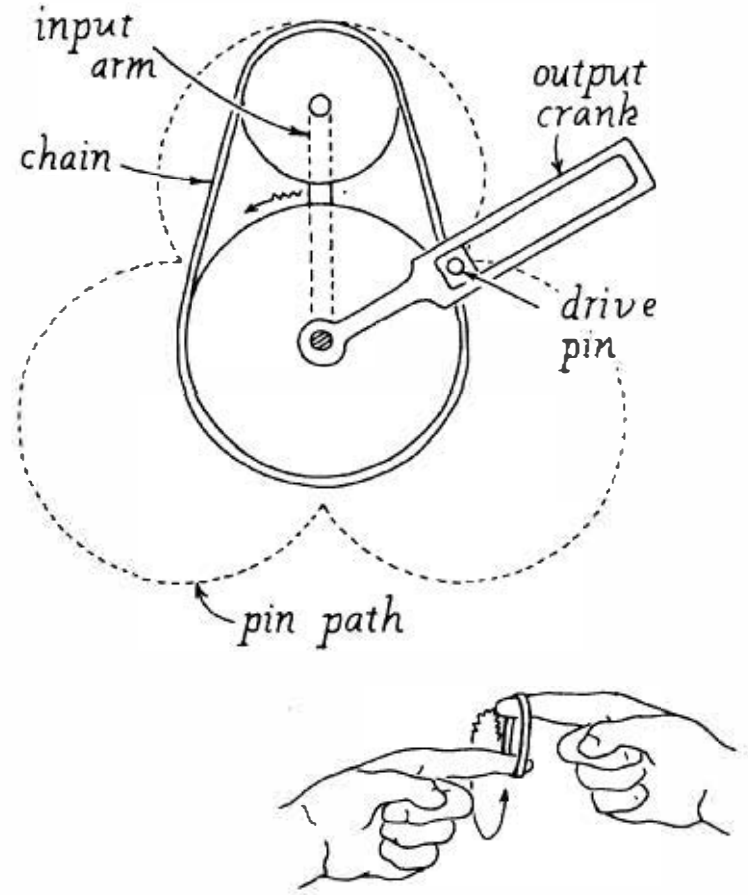
There is probably little or no slip between drive cam and input roller, but even if there is, the length of the output step is not affected since this is determined by the trigger system and by the sequencing of the control solenoid (slip would, of course, increase the wear rate of the system).

Figure 15–9 shows another roll cam system. This is similar to the first, but the friction roller is cylindrical rather than eccentric, and the output can be stopped in any one of ten positions rather than in just one. Kinetic energy in the stopping driven member rocks the support member to disengage input and output rolls when the stop arm catches the stop wheel.

Chain and Belt Drive

All of the mechanisms discussed so far have used only rigid links (unless operating speeds were high enough to make them act as elastic members). Some systems will now be considered in which flexible links, such as chains and belts, are used. Since such members have some built-in elasticity, they often act to cushion shocks and impacts in a desirable manner. It must be remembered, of course, that any impact damper which returns energy to the system must return it at the right time (between drive cycles rather than on top of the next drive cycle) if it is, indeed, to reduce and not intensify, impact and shock effects. As long as this possible trouble is kept in mind, belts—and to a lesser extent—chains, can be used to cushion sudden starts and stops.

One simple system incorporating a chain is shown in Fig. 15-10. This system bears a strong family resemblance to an epicycloidal gear arrangement (Fig. 10–16) but is different in performance. The input member is a rotating arm, the output member a slotted crank. This crank is driven by a drive pin as with an epicycloidal gear system. Instead of the drive pin being mounted on a planet gear, however, it is now mounted on a chain which connects the freely rotating "planet pulley" and the stationary "sun pulley." The output crank in this system dwells for approximately 120 degrees of rotation of the input arm while the planet pulley is "walking along the belt." This action can be demonstrated by walking an elastic band around one index finger with



Drawing courtesy of PRODUCT ENGINEERING Magazine; Oct. 26, 1964; pp. 109, 110

Fig. 15–10. Chain and slider indexing system. The chain is "walked around" the fixed sun pulley by the motion of the planet pulley, periodically indexing the chain mounted drive pin.

the other, as in the illustration. Mark one point on the rubber band with ink or a piece of tape so you can see how the drive pin is periodically lifted and moved over the surface of the large stationary center pulley.

Figure 15–11 shows another belt system; this one involving a four-bar linkage and two timing belts. The input crank rotates about the input shaft (*16*), carrying with it a shaft (*20*) and pulley (*25*), that

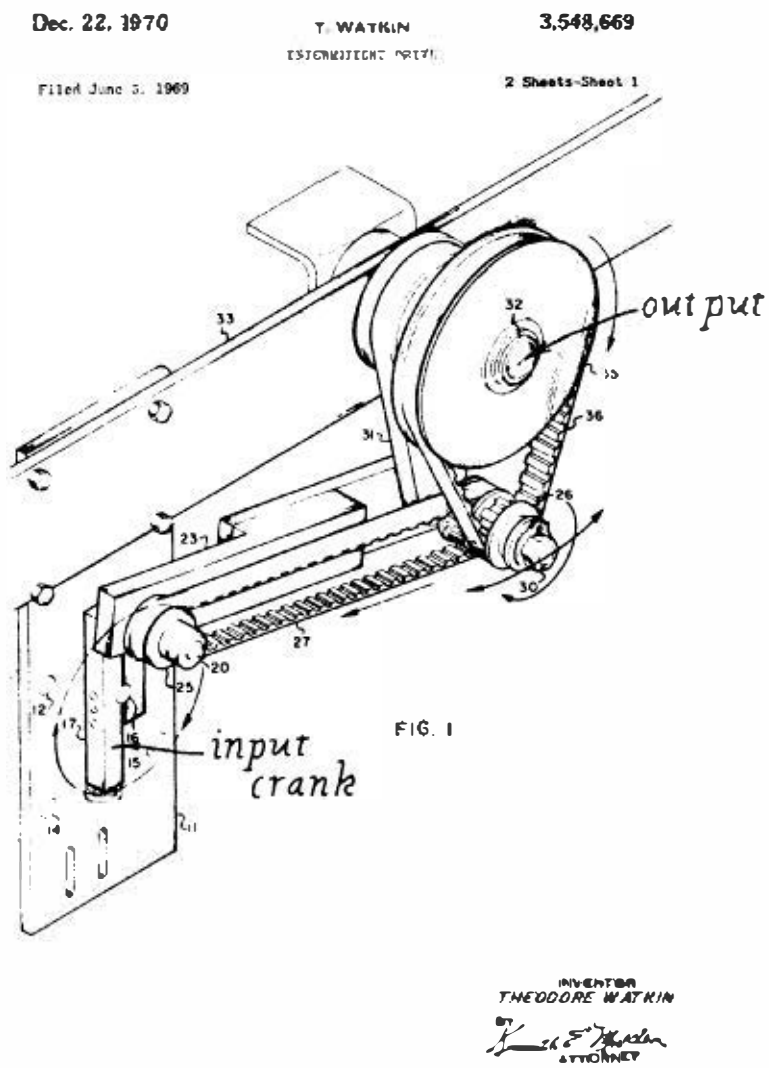


Fig. 15–11. Timing belt and four-bar linkage indexing mechanism. In this device, an oscillatory motion is added to a continuous motion to produce intermittent motion with instantaneous dwell. (U.S. Patent 3,548,669; T. Watkin.)

are not allowed to rotate with respect to the input crank. As the crank rotates, therefore, this "fixed" input pulley will tend to rotate the output pulleys. As the input crank rotates, however, it also oscillates the four-bar linkage (consisting of members *17*, *23*, *31*, and the frame). This oscillation tends to produce an output motion which periodically adds to and subtracts from the rotational motion produced by the input timing belt. Dual pulley (*26*) sometimes merely "walks along" output timing belt (*36*), and

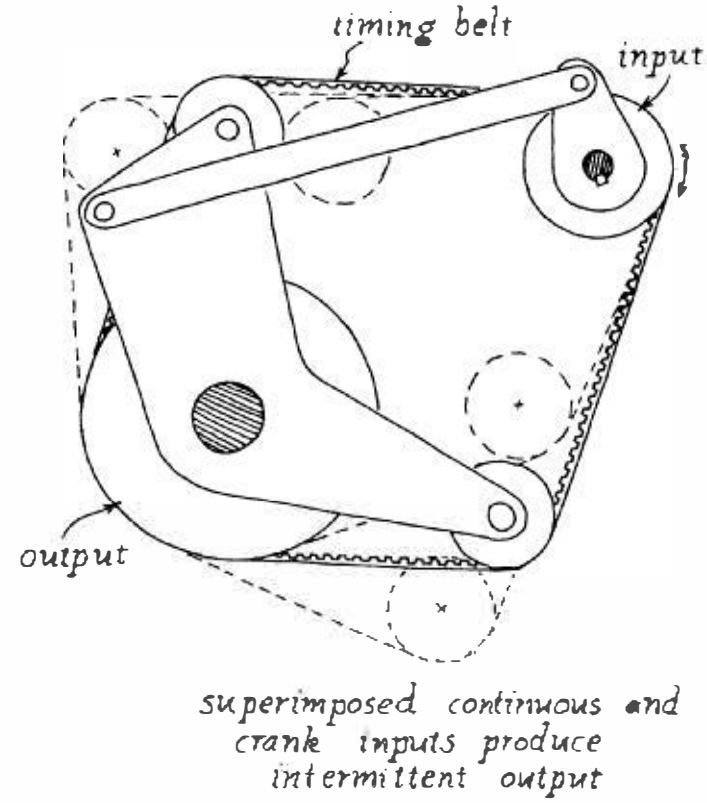
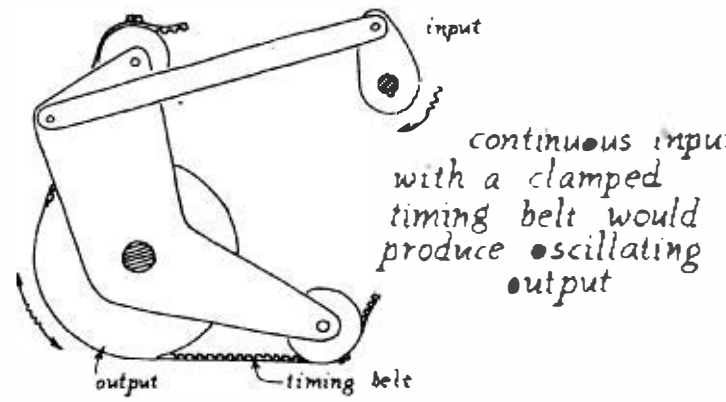


Fig 15–12. Another timing belt and linkage system. (See Fig. 15–13 for an explanation of how it functions.) Again, an oscillatory motion is added to and subtracted from a rotational motion.



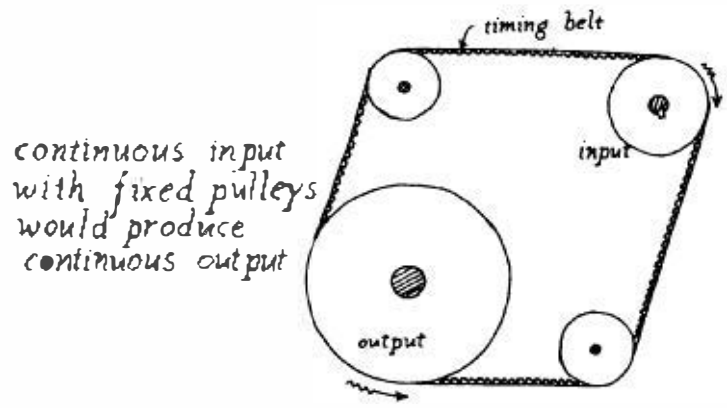


Fig. 15–13. Showing how the mechanism of Fig. 15–12 combines oscillatory motion with rotating motion to produce intermittent motion.

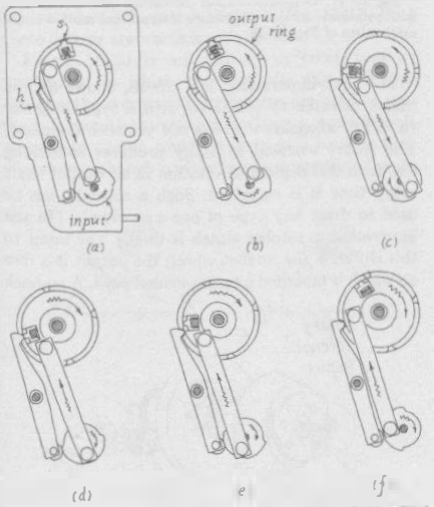
at oth times ives it. The result intermittent motion in shaft (32)

similar belt and crank arrangement is shown in Figs. 15–12 and 15–13. Again, the oscillatory motion of a linkage is add tinuous rotational motion that n in Fig. 15–13. A similar addition of tional and a reciprocal motion was shown in 9.

A system of the type shown in Fig. 15–11 theoretically produces only instantaneous dwell periods with a more or less sinusoidal output motion in-between. Some designers, therefore, have combined such systems with clutches and brakes to extend dwell periods. The clutch-brake unit is used to turn the input to the b dwell periods minimizing starting and stopping accelerations. The length of period w then be unde the control external computing device

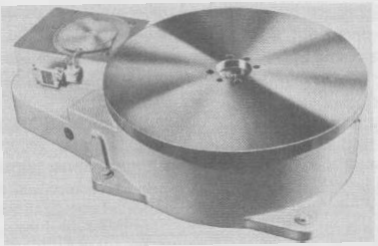
Linkage Drives

The versatile four-bar linkage can also be used other ways to produce intermittent rotary motion.



Drawing courtesy of the Jackson Machine Products Div., Power Company

Linkage ... is a spring loaded drive pin. Arm 'h', connected to input cam, periodically holds s the output ring to which the table is fastened.



Photograph courtesy of the Jackson Machine Products Div., Jack... Power Company

5. Indexing table with the drive system shown in Fig. 15–14.

Figure 15–14 shows the sequence of of a mech used in a machine tool gt able An input cam drives a linkage arm turn, rotates a drum w e indexing table. output ring, connected to the working surface of t table, surrounds this drum but is kept from moving by a locking lever, *h*, which is controlled by the input cam. A spring-loaded tooth, *s*, in the drum also engages this output ring

When the cam releases th ver, *h*, the spring-loaded tooth, *s*, can carry the output ring along with the drum as shown in (b) and (c) in t tion. At the end of th stroke, the locking lever depresses the spring-loaded tooth, as shown in (d), so that the drive drum and tooth can return to the starting position (e) and (f). The output ring dwells during port of the cycle Eventually, the drive tooth picks up the ring again and another index motion occurs. The complete indexing table is shown in Fig. 15–15.

Figure linkage n. This syste gear devices considered illa- tory motion with produ intermittent motion with inst ell periods. In fact, this linkage system is a close cousi of the gear system shown in Fig. 10–28 which d at some detail in the accompanying text

Miscellaneous

The remaining illustrations in this chapter show other ways to produce i ermittent mot Some were designed for special situat and may never find application again, but have been included, never-

theless, to make this book as complete as possible. Perhaps one of these unusual solutions will be the only possible solution for some future reader.

Other devices in this collection are included here simply because they cannot be classified with any of the devices which have gone before. It is quite possible that they may find widespread application in the future, and, indeed, are used rather broadly at the present time. The fact that they are presented last should not in any way indicate that they are least effective.

July 1, 1969 J. M. STEINKE 3,452,613

INTERMITTENT ROTARY MOTION TRANSMISSION

Filed Jan. 2, 1968 Sheet 2 of 3

FIG. 2

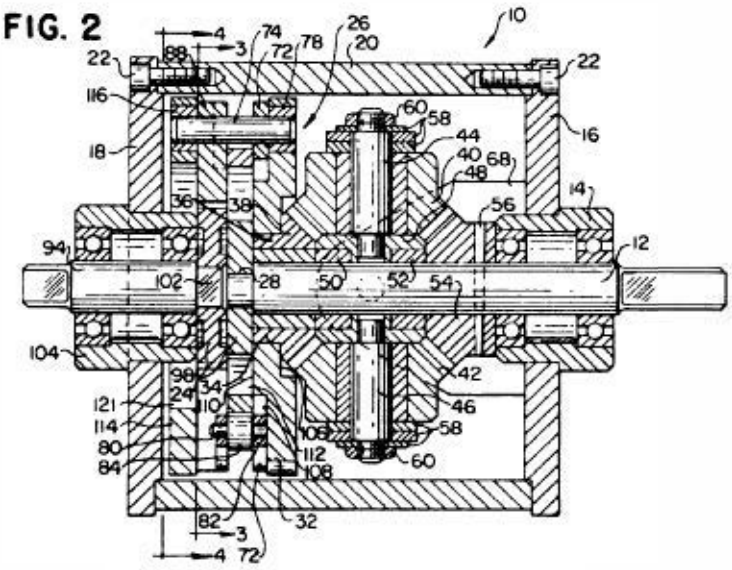
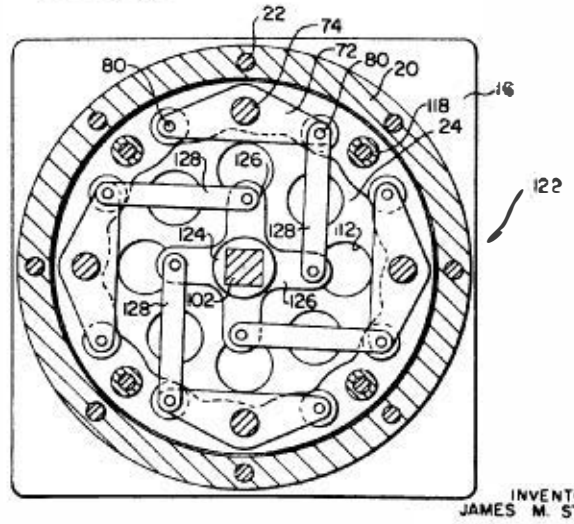


FIG. 5



INVENTOR
JAMES M. STEINKE
BY Louis A. Kline
Albert L. Sinclair
Spencer Green
HIS ATTORNEYS

Fig. 15-16. Cam and linkage indexing system. See Fig. 15-17 for description of operation. See also Fig. 10-28. (U.S. Patent 3,452,613; J. M. Steinke.)

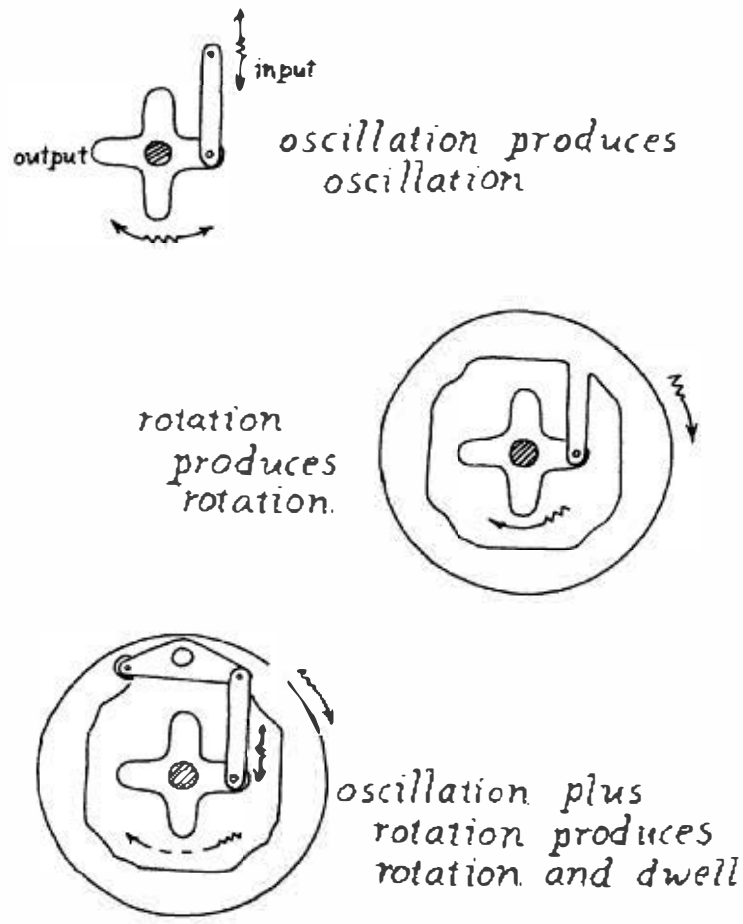
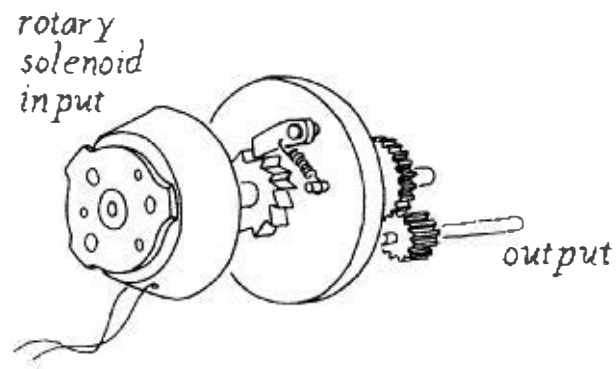


Fig. 15-17. Demonstration of oscillatory motion combined with rotational motion to ptoduce intermittent motion in the mechanism of Fig. 15-16.

The first illustration, Fig. 15-18, shows an old friend (a ratchet) combined with a type of driver (a rotary solenoid) that has not yet been discussed. The rotary solenoid typically produces something less than 360 degrees of rotation in an output shaft every time it is energized. Such a solenoid can be used to drive any type of one-way clutch. (In the illustration, a ratchet clutch is used.) The input to this clutch is the ratchet wheel; the output is a disc on which is mounted a spring-loaded pawl. A no-back



Drawing courtesy of MACHINE DESIGN Magazine, Nov. 21, 1968

Fig. 15-18. Rotary solenoid and ratchet clutch. The rotary solenoid typically produces something less than 360 degrees of rotation in an output shaft every time it is energized.

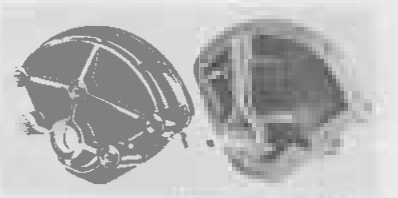


Fig. 15–19. Pneumatic vane type actuator with cover removed. This actuator produces 90 degrees of rotation in an output shaft whenever high-pressure air is introduced on the proper side of the vane.

pawl (not shown) would also be required. The following illustration Fig. 15–19, shows a pneumatic driver that can do the same job as the rotary solenoid but with higher torque. This vane type actuator produces 90 degrees of rotation in an output shaft whenever high-pressure air is introduced on the proper side of the vane. The device, then, can serve as a pneumatic "rotary solenoid" in conjunction with one-way clutches to provide the pneumatic equivalent of the system shown in Fig. 15–18. Both these drivers are worth knowing about.

Figure 15–20 is an entirely different type of mechanism in which two basically linear devices (pneumatic or hydraulic cylinders) are used to produce intermittent rotary motion. A considerable amount of torque could be developed by this type of system, although positioning accuracy would probably not be very great. Linear actuators of any kind could be used in a similar fashion. The output dwells whenever both drivers are at rest (extended

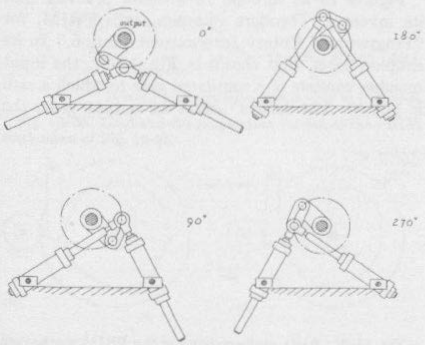
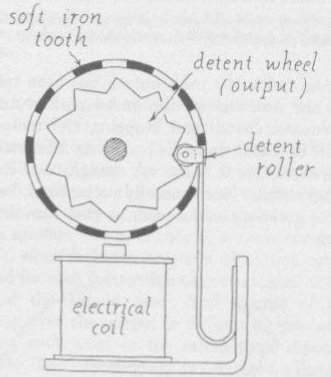


Fig. 15–20. Two pneumatic or hydraulic cylinders are operated in sequence to provide four-step output motion. (U.S. Patent 3,550,477; R. Kinnicutt, Jr.)

The device shown in Fig 15–21 bears some relationship to an inverse escapement and some to a ing motor, but not enough to either one to warrant in those chapters. It is one of the devices that have been invented to solve some special intermittent motion problem. The output wheel has ten soft-iron teeth and a ten-toothed, star-shaped detent wheel. When the electrical coil (the only input here) is energized, it will

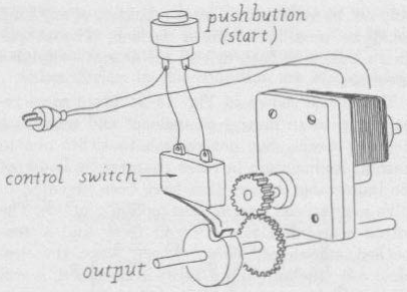


Drawing courtesy of MACHINE DESIGN Magazine; Dec. 23, 1965; p. 121 ff

Fig. 15–21. Electro-mechanical stepping motor. The output wheel has ten soft-iron teeth and a ten-toothed star-shaped detent wheel.

attract the nearest soft-iron tooth. When the coil is de-energized, the detent roller will move the output wheel a few degrees further. Periodic excitations of the electrical coil, therefore, will step the output wheel.

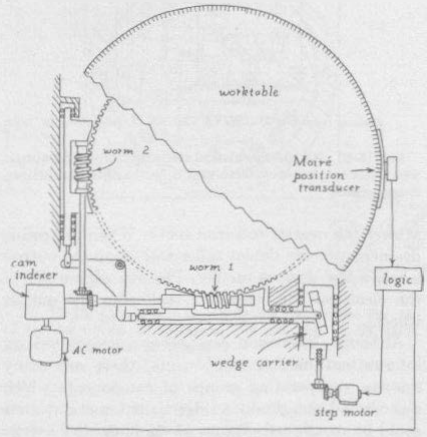
Although discussion has been concentrated on intermittent motion components, there are many systems (cooperating groups of components which alone could not produce intermittent motion) that could be mentioned. Figure 15–22 illustrates a very simple one, the switched electrical motor system. A push button is depressed momentarily to start the electrical motor. The first few degrees of rotation of the output shaft closes the control switch which is in parallel with the push button. This keeps the motor



Drawing courtesy of MACHINE DESIGN Magazine; Nov. 21, 196

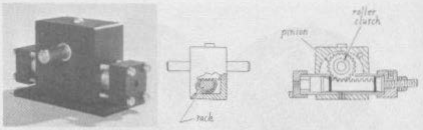
Fig. 15–22. Switched electrical motor. The push button is depressed momentarily to start the electrical motor.

running (after the push button has been released) until the cam reaches the [illegible] position, opening the control switch and stopping the motor. The motor, of course, must be braked by friction and, or the load, before it coasts far enough to re-close the control switch. This is usually no problem, however, with a gearhead motor such as that shown in Fig. [illegible]



Drawing courtesy of The Gleason Works

Fig. 15–23. Curvic coupling system. An AC motor operating through a cam indexing mechanism drives a pair of worm gears to position a work table near a desired position. A stepping motor operates a wedge cam system to improve the accuracy of the final position.



Drawing and photograph courtesy of Rotomation Inc.

Fig. 15–24. (*Left*) External view of a hydraulic-pneumatic stepper. (*Right*) Schematic of the interior mechanism.

Figure 15–23 shows the Curvic coupling system, a very sophisticated system that is used for high-performance machine tools. An AC motor operating through a cam indexing mechanism drives a pair of worm gears to [illegible] a work table near a desired position. A stepping motor operates a wedge cam s[illegible] to [illegible] the accuracy of the final position. A Moiré transducer (see Chapter 6) measures table mot[illegible] very [illegible] circuits [illegible] the [illegible] and actual table positions and control the two [illegible]e motors to produce the [illegible] motion.

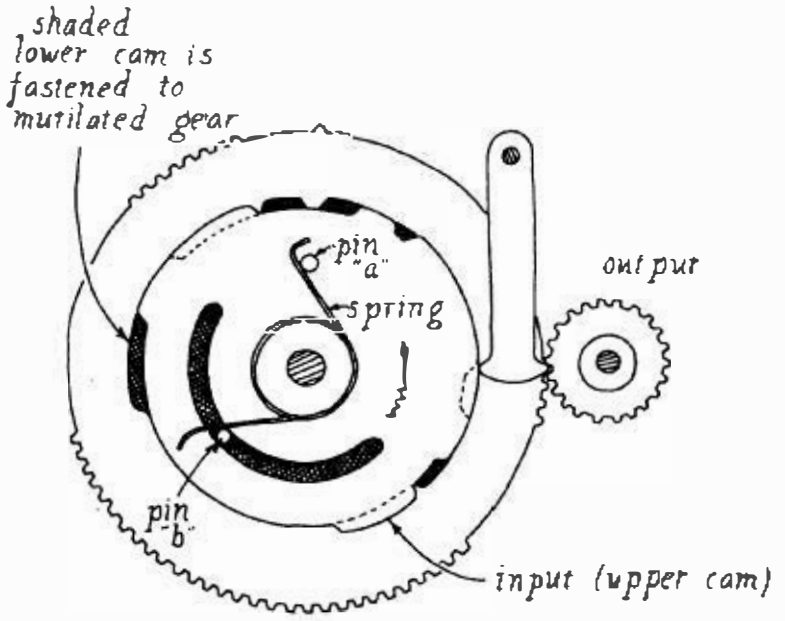
The hydraulic-pneumatic stepper is an interesting device using a one-way clutch to produce intermittent mot[illegible]; it is shown in Fig. 15–24. A hydraulic or pneumatic cylinder drives a rack which drives a [illegible]. A one-way roller clutch connects the [illegible] to the output shaft. The shaft, therefore, is indexed as the piston is moved back and forth. The degree of output mot[illegible] the stroke of the piston rack assembly

[illegible]

Figures 15–25 through 15–27 show a device that its inventor, Theodore Simpson, calls PRIM, for "Programmed Rotary Intermittent Mot[illegible]." In its simplest form, that shown in Fig. 15–25, the input member consists of a mu[illegible]ed gear to which a cam is pinned. The former drives an output gear, the

Fig. 15–25. Basic configuration for the PRIM mechanism (A) PRIM mechanism with the input in motion and the output at rest. (B) Output gear in motion. (U.S. Patent 3,442,146; T. Simpson.)

latter controls a locking lever (Fig. 15-25A) which releases the output gear (Fig. 15-25B) just prior to each drive cycle (and locks it again just after each motion cycle). In its basic form, the device bears a family resemblance to the detented mutilated gear system shown in Fig. 10-11, although the output is under better control in Mr. Simpson's design. The detent lever is spring loaded against the input cam, or is driven out of engagement with the output gear by the motion of that gear whenever the input cam allows this. Furthermore, additional refinements have been evolved. In Fig. 15-26 input is provided by one control cam which is spring loaded to the original cam (still fastened to a mutilated gear). The locking lever engages both cams and the output gear is always locked—no output will occur—until the two cams are aligned, i.e., a slot on each so that the locking lever can swing to the left. Pin, *a*, on the input cam loads a drive spring to drive the mutilated gear and lower cam through pin *b*, which operates in a slot in the input cam. The result is to combine



Drawings made from information supplied by T. Simpson

Fig. 15-26. Load-and-fire snap-action version of the PRIM mechanism of Fig. 15-25.

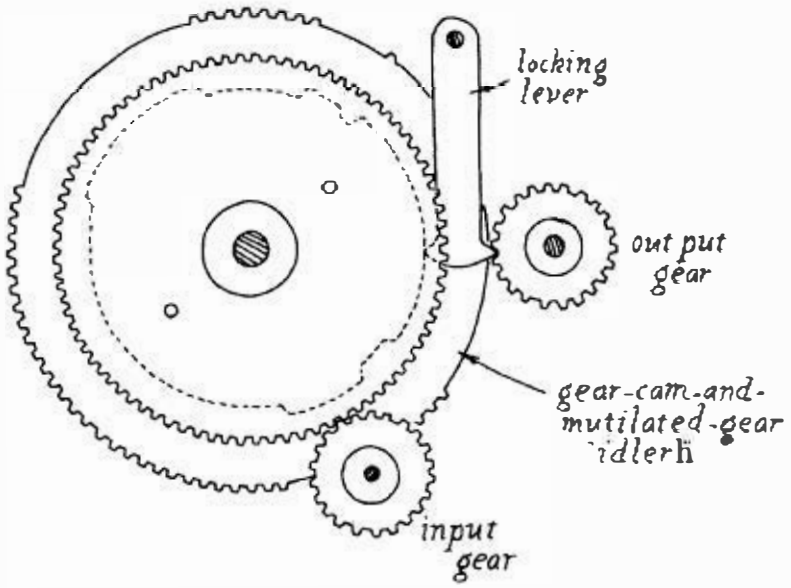


Fig. 15-27. In this version of the PRIM mechanism the input gear rotates three times for each revolution of the mutilated gear.

the features of a load-and-fire escapement (see Fig. 12-11) with the features of a mutilated gear.

The inventor also has pointed out (Fig. 15-27), that if either of these devices is driven through a step-down gear ratio the amount of output motion (or dwell) can vary from one full revolution of the input to another (with, of course, a cyclic repeating pattern), since different amounts of output can be produced for each interaction between the mutilated gear and the output gear. This degree of programming over the output is difficult or impossible to obtain with most of the mechanisms discussed previously. The various gears can be sized, of course, so that only two shafts are involved, making a very compact mechanism. Three shafts have been shown in Fig. 15-27 merely to clarify the operation of the design. As a result of this and other refinements, Mr. Simpson's device truly deserves the name programmed rotary intermittent motion. It would have the normal disadvantages of intermittent gearing at high speeds or heavy loads, but properly applied it should be quite a versatile and useful mechanism.

CHAPTER 16

Design of Intermittent Motion Mechanisms

PART I—DESIGN PROCEDURE

Now that there is a repertoire of mechanisms to choose from, let us see what steps should be taken to solve specific design problems. Since design is (at least, in part) a creative act it is a little arrogant, perhaps, to attempt to list a design procedure. I have known many successful designers whose approach differed considerably from that given below. Nevertheless, this procedure has worked for me, and presumably, it would also work for others.

1. Defining the Primary Problem

Never start to solve a problem until it has been clearly defined. In the case of indexing mechanisms, the primary problem is to produce intermittent motion of a certain type. The following are the basic specifications to this end:

a. Indexing rate required (number of steps per minute)
b. Indexing accuracy required
c. Dwell-motion pattern required
d. Size of load (heavy, medium, or lightduty)
e. Cost situation (an expensive mechanism is acceptable, or medium or low-cost must be achieved).

2. Defining Secondary Problems

A definition of indexing rate, load, cost, etc., usually determines which major classifications of indexing devices are candidates for a particular design solution. But these specifications alone rarely define the complete problem. Perhaps a size or weight restriction must be placed upon the indexing device. Perhaps special acceleration patterns are needed, or a "mechanical advantage" may be necessary. There may be unusual environmental problems; automatic vending equipment located at the seashore is exposed to a remarkable amount of salt content in the air, for example. In Table 16-1 are listed some of the many factors which the designer may have to consider in selecting a particular intermittent motion mechanism for a specific design.

3. Listing Primary Candidates

Next, consider the various types of intermittent motion mechanisms which have been discussed. Make a mental or written list of those which appear to be the most likely candidates for solving the primary problems (for example, a certain indexing rate, dwell-motion pattern, load capacity, etc.). Table 16-2, at the end of this section, lists the principal features of many of the devices which have been discussed.

The table defines the limits of my own experience with various types of intermittent motion mechanisms. Do not let my knowledge limit you, however, I am sure it would be possible to find some other designs which would be better (or worse) than those indicated here.

Table 16-3 gives detailed information on the input-output motion characteristics of most of

Table 16–1. Design Parameters

SPEED
- Cycling rate
- Velocity while in motion

SIZE

LOAD CAPACITY

COST

ENVIRONMENT
- Temperature
- Shock and vibration
- Vacuum
- Humidity
- Salt spray
- Dust
- Explosive

TYPE OF INPUT
- Continuous
- Impulse

DWELL AND MOTION PATTERN
- Duration of output dwell versus input motion
- Output displacement versus input displacement
- Number of dwells per revolution of output
- Number of output dwells per revolution of input
- Dwell-motion ratio

INPUT-OUTPUT
- Parallel or at right angles

STABILITY OF PERFORMANCE
- When new
- When worn

EASE OF CONTROL
- Mechanism stands alone
- Other components required: electronic drive circuits, sensors, hydraulic circuits, etc.

MOTION CURVES
- Involve impact
- Do not involve impact
- Can be altered by the designer

SYNCHRONOUS OR ASYNCHRONOUS
- Designed to operate at a fixed or variable cycling rate.

SERVICEABILITY

LEGAL REQUIREMENTS

- Accuracy
- Safety
- Noise

MECHANICAL ADVANTAGE
- Required
- Useful
- Not pertinent

DEGREE OF CONTROL OVER LOAD
- Good
- Fair
- Poor

RELIABILITY
- Will mechanism "always" function?

- What if it does not? Minor nuisance or catastrophe?

RISK QUOTIENT
- Does success depend on unknowns?
- Is this risk really necessary? Is there an alternate solution that involves less risk?

- What can be done to reduce or evaluate the unknowns?

EXPECTED PERFORMANCE VERSUS THAT OF THE

COMPETITION

FITTING COMPANY'S ABILITY
- Does it fit company's present or anticipated skills? In:
  - Design
  - Manufacturing
  - Sales
  - Service

POWER CONSUMPTION AND EFFICIENCY

UNUSUAL SALES FEATURES REQUIRED

CAPACITY TO TOLERATE OVERLOAD

OPERATING COSTS

WEIGHT

MANUFACTURE VERSUS PURCHASE

PATENT SITUATION
- Patentable
- Interference problems

the mechanisms illustrated in this text, and Table 16–4 lists some of the typical applications for various types of mechanisms. These three tables should be helpful in rapidly selecting possible candidates to solve your design problems.

4. Reviewing the List with the Secondary Problems in Mind

Getting a little deeper into the problem, you should attempt to eliminate any of the primary

candidates that fail to meet the important special or secondary requirements for the design. But care must be taken with all of this "eliminating," as the solution to your design problem may very well consist of a slightly modified mechanism. Do not discard a category merely because the obvious members of that branch of the "family" fail to meet one of the requirements. Again, Table 16-2 reflects my own knowledge or experience in this field; do not feel limited or confined by it.

5. Choose the Most Promising Approach

Perhaps only one "solution" will be found by following the previous steps; or you may have found several possibilities. In any event, the time has come for you to select one candidate for further evaluation. This could be either a "pure" mechanism (for example, an external Geneva), orat might be a modified mechanism to satisfy a special situation (for example, a Geneva driven by a four-bar linkage to modify the acceleration pattern). It also may be a combination of two or more types of intermittent motion devices (for example, a Geneva driven by a clutchbrake combination to modify the dwell-motion ratio).

In order to make a selection, you will probably be facing the almost inevitable problem of choosing between apples and hammers. If it were a decision between apples and oranges, it might not be so difficult, for at least they are both pieces of fruit. But in most design situations, there will be two or more possible solutions to a design problem, each having an entirely different set of advantages and disadvantages. One solution, for example, may offer everything you need in the way of reliability, but the mechanism may be a little larger than you plan, and its cost is on the high side. A second solution may offer exceptional speed, a high degree of patent protection, and be easy to make on the company's molding machines. A third solution may be very quiet, remarkably easy to service, and have almost infinite life. Presumably, all three possibilities satisfy your main requirements of indexing speed, load capacity, etc. Your job then is to decide whether low noise level is more desirable than patent protection or unusual reliability, for example. These are the kinds of questions that make a designer yearn for the wisdom of Solomon, for in any event, you must make a selection.

6. Incorporating the Mechanism into Your Design

You have now picked the most promising candidate. Presumably, it is only a component of a larger machine or instrument. Next, you must incorporate it into that larger design. As you do this, you will almost inevitably encounter major or minor problems which you bad not anticipated. The size is wrong; or the mechanism requires right-angle shafts and yet your design would be neater if input and output were on parallel shafts; or the device will produce impact and you find you cannot incorporate the large bearings required to tolerate this; etc.

Do not give up on encountering these obstacles, overcome them as well as possible and continue until you strike an obstacle that is definitely insurmountable or until the mechanism is finally successfully incorporated into your design, even though it may be at some sacrifice to your original intentions. This is not recommended simply because I think designers should be stubborn, but because I think a struggle of this kind is often necessary to uncover the *real* problems in a given design situation. At the beginning of your work, you may think you know what the problems are, but it is only after struggling through a complete layout that you really uncover them all.

Do not forget to consider what the performance of the design will be after it has worn a little, by using drawings or plastic cutouts, or models, to study the effects of dimensional changes. Remember also, that intermittent motion places a severe burden on most mechanisms and small dimensional changes can cause significant changes in performance. Try to discover and eliminate all the potential wear points that might affect performance. Also consider all the design parameters listed in Table 16-1 and honestly list the advantages and disadvantages of your new system, now that you have worked it out. This is also a good time to calculate a first, rough cost estimate to compare with your initial estimate; are you in the ball park or did the modifications you were forced to make prove too costly?

7. Reconsidering Other Possible Solutions

Now that you know what the problems really are, reconsider some of the other possible solutions. You

may see one that on second thought is quite good and gets around some of the unexpected difficulties encountered while incorporating your first choice into the design. If you make another selection at this point, carry it all the way, also. You may uncover another crop of secondary problems or you may find that it is, indeed, an improvement over your first choice.

8. Warning

Try not to fall into the designer's main trap which is picking a particular solution simply because he thought of it, ego-satisfying though this may be. Pick the best solution for this design even if it happens to be the most common one available; proving that there is at least one designer who can place the welfare of his customer and his company above any desire for self-expression.

9. Design Refinements

If your second solution encounters a new crop of difficulties, you may have to go through the cycle a third time. But eventually you will find one that appears to be an acceptable answer to your main and secondary problems. Go still further; evaluate the "final" design to see if shapes, assemblies, motions, or controls can be simplified even more. Can you expand or reduce function? Can you combine elements? Can you optimize the sequence of operations? Can you improve safety? In other words; act objectively as a design review team, to put those final touches on the device; all of which separates the master designer from the "couldn't care less" school.

It is not to be assumed that your design should be improved indefinitely; nothing will drive a design boss up the wall faster and for better reason. A design that is "too good" is almost as uneconomical as one that is not good enough. The idea is not to let your initial "final" solution rest until you have taken a quick look, at least, at improvements of the type described.

10. Design Review

The time has now come to turn your design over to others for their evaluation. The average newspaper reader can easily detect any bias that the reporter would swear does not exist in his prize article. By the same token, another designer or engineer can spot deficiencies in a design that the designer simply cannot see perhaps because of personal prejudice based on his own experience; because of his desire for self-expression; or because he has just plain overlooked something; etc. Let someone else evaluate your design even if your company does not have a formal design review group. And listen to your critic--debate his conclusions to learn what they are based upon, then modify your design or not, depending upon whether his observations seem valid. But do not be on the defensive until you have really understood what he is saying and have determined whether or not he has actually uncovered a valid problem.

This is also a good time to get another cost estimate. But this time it should be done by someone else, and should be as rigorous as time and information will allow.

By the time you get through all this, you may find that your solution is no longer a solution and you will have to start all over. If so, go back to Steps 3 and 4, or possibly even 1 and 2, and repeat the entire process. Each time through will make you more aware of the total problem and of the advantages and disadvantages of the various solutions in solving that problem. Not infrequently, you will have to throw away all possible common solutions and innovate a new intermittent motion mechanism or system to solve your particular problem. If this were not the case, this book would have been only a fraction of its present length! As long as you are innovating because the design demands it, rather than because your ego demands it, no one can complain. In fact, this is what you are paid for.

Well, we have come a long way and, hopefully, know a great deal more about the mechanics and mechanisms of intermittent motion. This class of mechanism has had a long and important past, as the "Historical Notes" have suggested. I hope each of you will contribute something to its endless and important future.

Design Parameters

Some of the factors which must be considered in selecting a particular intermittent motion mechanism for a particular design are listed in Table 16–1. But the characteristics of the mechanisms we have discussed and a partial listing of their applicability are given in Tables 16-2, 16-3, and 16–4.

Table 16-2. Specifications and Characteristics of Various Types of Intermittent Motion Mechanisms

Type of Mechanism	Cycling Rate (SPM)	Dwell-Motion Ratio (Output-) Time	Relative Load Capacity	Relative Cost	Indexing Precision	Performance Stability		Controls for Mechanisms are:	Degree of Control Over Load
						New	Worn		
Impulse ratchet	1500	High	Low	Low	Moderate	Fair	Fair to poor	Simple	Fair
Cam ratchet	A few hundred	Moderate	Moderate to high	Low to moderate	Moderate	Good	Fair	Simple	Fair to good
Cam	1000	Moderate	Very high	Moderate	High	Excellent	Good	None	Excellent
Instrument Genevas (external)	10,000	Low	Low	Very low	Moderate	Good	Fair	None	Good
Machine Geneva (external)	Hundreds	Low	High	Moderate	Moderate	Excellent	Good	None required	Good
Mutilated gearing	5000	Low to Moderate	Very low	Very low	Moderate	Good	Good	None required	Good
Cycloidal gearing	Few thousand	Very low	Low to moderate	Moderate	Poor to fair	Good	Fair	None required	Excellent
Differential gearing	Few thousand	Very low	High	Moderate to high	Poor to fair	Excellent	Good	Moderate complexity	Excellent
Clock and watch escapements (tuned)	10 to 100	Low	Very low	Low to moderate	Very high	Excellent	Poor	None	Excellent
Machine escapement	10,000	Moderate to high	Low	Low	High	Fair	Fair	Simple	Fair
Inverse escapement	3500	High	Low	Very low	Moderate	Fair	Poor	None required	Fair
Clutch-brake systems	10,000	Moderate to high	Moderate to high	Moderate to high	Low to moderate	Fair	Fair	Complex	Fair to good
Step motor	960,000	Low to high (depending on type)	Low to high (depending on type)	Moderate to high	Moderate	Fair	Poor	Complex	Fair to good
Star wheel	Hundreds to thousands	Moderate	High	Low to moderate	Moderate	Excellent	Good	None required	Good
Roll cam	Thousands	Moderate to high	Moderate to high	Moderate	Moderate	Good	Poor	Moderate complexity	Fair to good

Type of Mechanism	Type of Input to Mechanism	Input Displacement		Output Displacement per Motion	Number of Output Dwells		Designed to Run at Fixed (Synchronous) or Variable (Asynchronous) Rate	Input-Output Parallel or Right Angle
		During Output Dwell	During Output Motion		Per Output Revolution	Per Input Revolution		
Impulse ratchet	Electrical pulses or cam loaded spring	5° to 30°	5° to 30°	1° to 45°	8 to 360	D.A.	Either	D.A.
Cam ratchet	Rotating cam	90° to 300°	60° to 270°	0.1° to 90°	4 to hundreds	1 to 2	Fixed	Parallel
Cam	Rotating shaft	90° to 300°	90° to 270°	90° to 270°	1 to 4	1 to 5	Fixed	Either
Instrument Geneevas (external)	Rotating shaft	200° to 270°	100° to 90°	20° to 90°	4 to 8	1 to 2	Fixed	Parallel
Machine Geneva (external)	Rotating shaft	200° to 270°	160° to 90°	20° to 90°	4 to 8	1 to 2	Fixed	Parallel
Mutilated gearing	Rotating shaft	10° to 350°	10° to 350°	10° to many turns	1 to 36	1 to 5	Fixed	Parallel
Cycloidal gearing	Rotating shaft	Few degrees (theoretically 0°)	30° to 360°	30° to nearly 360°	1 to 12	1 to 12	Fixed	Parallel
Differential gearing	Two rotating shafts	0° to many turns	Fractions of a degree to many turns	Fraction of a degree to hundreds of degrees	One to hundreds	One to hundreds	Fixed	Either
Clock and watch escapements (tuned)	Stalled rotating shaft	0°	10° to 30°	10° to 30°	12 to 36	12 to 36	Fixed	Parallel
Machine escapement	Rotating shaft	Few degrees to many turns	Few degrees to many turns	Few degrees to many turns	1 to 100	Fraction of one to 100	Variable	Parallel
Inverse escapement	Rotating shaft or electrical pulses	5° to 300°	5° to 180°	4° to 60°	6 to 90	D.A.	Variable	Parallel
Clutch-brake systems	Rotating shaft	Partial revolution to many revolutions	Partial Revolution to many revolutions	A few degrees to many revolutions	One to hundreds	One to hundreds	Variable	Mostly parallel
Step motor	Electrical pulses	D.A.	D.A.	0.9° to 180°	2 to 400	D.A.	Either	D.A.
Star wheel	Rotating shaft	30° to 300°	60° to 320°	60° to 360°	1 to 8	1 to 3	Fixed	Parallel
Roll gear	Rotating shaft	180° to many turns	30° to many turns	30° to 360°	1 to 12	1 to 5	Variable	Parallel

Table 16–2 (Cont.). Specifications and Characteristics of Various Types of Intermittent Motion Mechanisms

Type of Mechanism	Reliability	Basic Mechanism Offers Mechanical Advantage	Related Components Required	Relative Size	Degree of Control Over:		Is Impact Present?	Is Jerk Present	Discussed in Chapter
					Motion Curves	Dwell-Motion Ratio			
Impulse ratchet	Good	No	Switch or pulse circuits	Small	Poor	High	Yes	Yes	7
Cam ratchet	Good to excellent	A little	None	Small to moderate	Fair	Poor	A little	Probably	7
Cam	Excellent	A little	None	Small to large	Excellent	Fair	No	Can be avoided	8
Instrument Geneva (external)	Good	No	None	Very small	None	None	Yes	Yes	9
Machine Geneva (external)	Excellent	No	None	Moderate	None	None	A little	Yes	9
Mutilated gearing	Fair to good	A little	None	Small	None	Fair	Yes	Yes	10
Cycloidal gearing	Fair to good	A little	None	Moderate	Poor	Poor	No	No	10
Differential gearing	Excellent	Yes	Clutches (sometimes)	Moderate to large	Poor	Poor	No	No	10
Clock and watch escapements (timed)	Excellent	Very high	None	Very small to small	None	None	Yes	Yes	11
Machine escapement	Good	Yes, high	Control solenoid & drive circuits or control shaft	Small	None	High	Yes, high	Yes	12
Inverse escalement	Fair	No	Drive circuits	Small	None	High	Yes	Yes	12
Clutch-brake systems	Fair to good	No	Control circuits	Moderate to large	Fair	High	With some types	With most types, yes	13
Step motor	Fair to good	No	Drive circuits	Moderate	Fair	High	No	No	14
Star wheel	Excellent	No	None	Small to moderate	Fair	Poor	A little	Yes	15
Roll cam	Good	Yes	Solenoid and drive circuits	Small to moderate	Good	High	No	No	15

Table 16–3. Input–Output Motions for Various Intermittent Motion Mechanisms

Mechanism	Input is:	Typical Input Angle During Output Dwell (Approx.)	Typical Input Angle During Output Motion (Approx.)	Typical Output Angle per Step (Approx.)	Example Shown in Figure:	Are Different-Length Steps Possible in One Output Revolution?
Variable stroke ratchet	Mech.	10ñ to 90ñ	10ñ to 90ñ	10ñ to 90ñ	7–12	Yes
Double action ratchet	Mech.	5°	15°	15ñ	7–13	No
Double action ratchet	Mech.	Few degrees	180°	33ñ	7–14	No
Reversible ratchet	Mech.	15ñ	15ñ	15ñ	7–16	No
Bi-directional ratchet	Electr.	10ñ	10ñ	30ñ	7-20	No
Multi-pawl ratchet	Mech.	2°	2°	2°	7–21b	No
Multi-pawl ratchet	Mech.	0.1ñ	0.1ñ	0.1ñ	7–23	No
Double output ratchet	Mech.	15ñ and 120ñ	15ñ and 15ñ	15° and 15ñ	7–24	No
Square ratchet	Pneumatic	Linear stroke	Linear stroke	90ñ	7-25	No
Ball ratchet	Mech.	45ñ	45ñ	45ñ	7–26	No
Electrical ratchet	Electr.	5°	5°	30ñ	7-27	No
Blade ratchet	Electr.	3°	3°	1.5°	7–28	No
Instrument ratchet	Electr.	10°	10ñ	36°	7–33	No
Solenoid ratchet	Electr.	Linear stroke	Linear stroke	36°	7–34	No
Linkage ratchet	Mech.	300°	60°	18°	7-35	No
Dual output linkage ratchet	Mech.	180°	180°	18ñ and 45ñ	7–37	No
Silent ratchet	Mech.	1ñ to 60ñ	1ñ to 60ñ	1ñ to 60ñ	7–40	Yes
Variable output ratchet	Mech.	21.8°	21.8°	5.4ñ to 21.8° (15 options)	7-41	Yes
Eccentric ratchet	Mech.	320ñ	40°	20ñ	7-42	No
Light-duty face cam	Mech.	0°	±5°	18ñ and 18ñ	8–2	No
Roller gear cam	Mech.	90ñ to 270ñ	90ñ to 270ñ	15ñ to 90ñ	8–7	No

Table 16–3 (Cont.). Input–Output Motions for Various Intermittent Motion Mechanisms

Mechanism	Input is:	Typical Input Angle During Output Dwell (Approx.)	Typical Input Angle During Output Motion (Approx.)	Typical Output Angle per Step (Approx.)	Example Shown in Figureh	Are Different-Length Steps Possible in One Output Revolution?
Roller input cam	Mech.	10h	170h	30°	8-10	No
Stationary cam	Mech.	324h 60°	36h 300h	36° 60°	8–11 8–12	Yes
Tschudi cam	Mech.	80°	280°	180°	8–13	No
Barrel cam	Mech.	90h to 270h	90h to 270h	90h to 18°	8–19	No
Reciprocating input cam	Mech.	Linear stroke	Linear stroke	72°	8–21	No
Conjugate cam	Mech.	180° to 00h	90° to 360°	45° to 360°	8-22 8–23	Yes
Helical cam	Mech.	180°	1260h	180°	8–25	No
Worm cam	Mech.	180°	180°	9°	8-26	No
Spring start cam	Mech.	90h	90h	90°	8-28	No
4-slot external Geneva	Mech.	270h	90h	90h	9–1	No
5-slot external Geneva	Mech.	252h	108°	72h	9–10	No
6-slot external Geneva	Mech.	240h	120h	60°	9–10	No
7-slot external Geneva	Mech.	$231\frac{1}{7}$°	$128\frac{4}{7}$°	$51\frac{3}{7}$°	None	No
8-slot external Geneva	Mech.	225°	135h	45h	9–10	No
10-slot external Geneva	Mech.	216h	144h	36°	9–10	No
12-slot external Geneva	Mech.	210°	150h	30°	None	No
14-slot external Geneva	Mech.	$205\frac{5}{7}$h	$154\frac{2}{7}$°	$25\frac{3}{7}$°	None	No
18-slot external Geneva	Mech.	200°	160h	20h	None	No
4-slot internal Geneva	Mech.	90h	270h	90h	9–2	No
5-slot internal Geneva	Mech.	108°	252h	72°	None	No
6-slot internal Geneva	Mech.	120h	240h	60°	None	No

Table 16–3 (Cont.). Input–Output Motions for Various Intermittent Motion Mechanisms

Mechanism	Input is:	Typical Input Angle During Output Dwell (Approx.)	Typical Input Angle During Output Motion (Approx.)	Typical Output Angle per Step (Approx.)	Example Shown in Figure:	Are Different-Length Steps Possible in One Output Revolution?
7-slot internal Geneva	Mech.	128½°	231½°	51½°	None	No
8-slot internal Geneva	Mech.	135°	225°	45°	9–11	No
10-slot internal Geneva	Mech.	144°	216°	36°	None	No
12-slot internal Geneva	Mech.	150°	210°	30°	9–11	No
14-slot internal Geneva	Mech.	154¼°	205¾°	25¾°	None	No
18-slot internal Geneva	Mech.	160°	200°	20°	None	No
4-slot spherical Geneva	Mech.	180°	180°	90°	9–3	No
6-slot spherical Geneva	Mech.	180°	180°	60°	None	No
8-slot spherical Geneva	Mech.	180°	180°	45°	None	No
Light-duty 6-slot external Geneva	Mech.	320°	40°	60°	9–15	No
Multi-roller 4-slot external Geneva	Mech.	60°	60°	90°	9–16	No
Variable dwell Geneva	Mech.	80° and 140° (alternates)	60°	90°	9–17	No
Chain drive Geneva	Mech.	90°	450°	90°	9–18	No
Slider crank Geneva	Mech.	240°	120°	90°	9–19	No
Hypocycloidal crank Geneva	Mech.	270°	90°	90°	9–20	No
Slider crank Geneva	Mech.	200°	160°	72°	9–24	No
Elliptical gear Geneva	Mech.	90°	90°	90°	9–27	No
Multilated gear	Mech.	350° to 10°	10° to 350°	10° to many turns	10–10	Yes
Long-tooth mutilated gear	Mech.	340°	380°	360°	10–13	Yes

Table 16–3 (Cont.). Input–Output Motions for Various Intermittent Motion Mechanisms

Mechanism	Input is:	Typical Input Angle During Output Dwell (Approx.)	Typical Input Angle During Output Motion (Approx.)	Typical Output Angle per Step (Approx.)	Example Shown in Figure:	Are Different-Length Steps Possible in One Output Revolution?
Mutilated gear counter mechanism	Mech.	324ñ	36°	36°	10-12	Yes
Multiple output mutilated gear	Mech.	60ñ and 60ñ	60ñ and 180ñ (alternates)	60ñ and 300ñ (alternates)	10-14	Yes
4:1 Hypocycloidal gear	Mech.	Theoretical 0°	90ñ	90ñ	10-3	No
8:1 Hypocycloidal gear	Mech.	Theoretical 0°	45°	45ñ	10-15	No
1:1 Epicycloidal gear	Mech.	Theoretical 0°	360ñ	360ñ	10-16	No
2:1 Epicycloidal gear	Mech.	Theoretical 0°	180°	180°	10-17	No
8:1 Epicycloidal gear	Mech.	Theoretical 0ñ	45°	45ñ	None	No
Grooved cam planet gear	Mech.	100ñ	260ñ	360°	10-18	Yes
Non-circular planetary gear	Mech.	60°	60°	60°	10-19	Yes
Eccentric hypocycloidal gear (non-rhtating)	Mech.	Theoretical 0°	360ñ	36ñ	10-20	No
Single input differential gear	Mech.	Theoretical 0°	180°	180°	10-24	No
Skewed block differential	Mech.	Theoretical 0°	Fraction of one, to many degrees	Fraction of one, to many degrees	10-26	No
Rack and idler gear	Mech.	180°	180°	90ñ	10-27	Yes
Segment gear	Mech.	Theoretical 0°	22½°	22½°	10-28	No
Three-gear system	Mech.	Theoretical 0ñ	360°	360ñ	10-31	No
Programmable gear	Mech.	20ñ to 340ñ	340ñ to 20°	40ñ to 720ñ	10-32	Yes
Verge escapement	Mech.	0°	16½ñ	16½ñ	11-8	No
Deadbeat escapement	Mech.	0°	13ñ	13ñ	11-13	No
Gravity arm escapement	Mech.	0°	120ñ	120ñ	11-15	No

Table 16–3 (Cont.). Input–Output Motions for Various Intermittent Motion Mechanisms

Mechanism	Input is:	Typical Input Angle During Output Dwell (Approx.)	Typical Input Angle During Output Motion (Approx.)	Typical Output Angle per Step (Approx.)	Example Shown in Figure:	Are Different-Length Steps Possible in One Output Revolution?
Tuned escapements	Mech.	0°	24°	24°	11–17	No
Simple machine escapement—two pallets	Mech. or Electr.	0°	10° to 180°	10° to 180°	12–1	Yes
Simple machine escapement—single pallet	Mech. or Electr.	0°	10° to 360°	10° to 360°	12–2	Yes
Disc escapement	Mech.	315°	45°	180°	12–6	No
Slip clutch escapements	Mech. and Electr.	Less than one, to many turns	A few degrees, to many turns	10° to many turns	12–7 12–9	Yes
Load-and-fire escapements	Mech. and Electr.	Less than one, to many turns	A few degrees	10° to 360°	12–11 through 12–15	Yes
Friction escapements	Mech.	One or more turns	720°	360°	12–17	Yes
Controlled output escapement	Mech.	300°	60°	60°	12–19	No
Variable stroke escapement	Mech.	0°	9° to 360°	9° to 360° (21 options)	12–21	Yes
Light-duty inverse escapement	Mech. or Electr.	5°	5°	18°	12–22 12–26 12–36	No
Very light-duty inverse escapement	Mech.	300°	60°	22½°	12–29	No
Heavy-duty cam-driven inverse escapement	Mech.	Few degrees	180°	4°	12–33	No
Heavy-duty, reciprocating inverse escapement	Mech.	Linear stroke	Linear stroke	20°	12–35	No
Spring clutch	Mech.	0° to many turns	10° to 360°	10° to 360°	13–2 13–18 13–28 13–29	Yes

Table 16–3 (Cont.). Input–Output Motions for Various Intermittent Motion Mechanisms

Mechanism	Input is:	Typical Input Angle During Output Dwell (Approx.)	Typical Input Angle During Output Motion (Approx.)	Typical Output Angle per Step (Approx.)	Example Shown in Figure:	Are Different-Length Steps Possible in One Output Revolution?
Low-cost electrical clutch (light-duty)	Mech.	0°, to many turns	90° to 360°	90° to 360°	13-4	Yes
Heavy-duty pawl clutch	Mech.	90°, to many turns	360°	360°	13-19	No
High-performance electrically actuated clutches	Mech.	Few degrees, to many turns	Few degrees, to many turns	Few degrees, to many turns	13-21 13-22 13-24 13-31 13-34	Yes
Overrunning clutches (roller, sprag, etc.)	Mech.	0°	Few degrees, to 360°	Few degrees, to 360°	13–18 13-25 13-26 13-27 13-30	Yes
Permanent magnet stepping motor	Electr.	—	—	45° to 180° (typical)	14-10 14-12	Yes
Variable reluctance stepping motor	Electr.	—	—	15° (typical)	14–16	Yes
Slo-Syn stepping motor	Electr.	—	—	0.9° 1.8°	14–13 14-14	Yes
Electro-hydraulic stepping motor	Electr.	—	—	1.5° and 3°	14–20 14-21	Yes
Flexible-spline stepping motor	Electr.	—	—	0.45°	14–22 14–23	Yes
Ratchet stepping motor	Electr.	—	—	10° to 45°	14–28	Yes
Planar stepping motor	Electr.	—	—	0.001″	14-24 through 14-27	Yes
One-stop external star wheel	Mech.	180°	180°	360°	15–4	No
Two-stop external star wheel	Mech.	90° or 270° or other	90°	180°	15-3	No
Three-stop external star wheel	Mech.	30° or 90° or 270°	90°	120°	15–4	No
Four-stop external star sheel	Mech.	30° or 90° or 270°	90°	90°	15-4	No

Table 16–3 (Cont.). Input–Output Motions for Various Intermittent Motion Mechanisms

Mechanism	Input is:	Typical Input Angle During Output Dwell (Approx.)	Typical Input Angle During Output Motion (Approx.)	Typical Output Angle per Step (Approx.)	Example Shown in Figure:	Are Different-Length Steps Possible in One Output Revolution?
Five-stop external star wheel	Mech.	48° or 108° or 288°	72°	72°	15-4	No
Six-stop external star wheel	Mech.	30° or 60° or 120° or 300°	60°	60°	15–4	No
One-stop internal star wheel	Mech.	60° to 180°	180° to 320°	360°	15–6	No
Two-stop internal star wheel	Mech.	90° or 270°	90°	180°	15–6	No
Three-stop internal star wheel	Mech.	60° or 180° or 300°	60°	120°	15–6	No
Roll cam	Mech.	Few degrees, to many turns	36° to 360°	36° to 360°	15–8 15–9	Yes
Pin-chain drive	Mech.	240°	120°	120°	15–10	No
Linkage-belt drive	Mech.	Theoretical 0°	180°	45° to 360°	15–11 15–12	No
Four-bar linkage	Mech.	180°	180°	90°	15–14	No
Rotary solenoid	Electr.	0°	10° to 180°	10° to 180°	15–18	No
Four-piston drive	Hydraulic or Pneumatic	Linear strokes	Linear strokes	90°	15–20	No
Gleason indexer	Mech. and Electr.	—	—	Small fractions of a degree, to 360°	15–23	Yes
Hydraulic rack actuator	Hydr.-Pneum.	Linear stroke	Linear stroke	30° to 360°	15–24	Yes
Basic PRIM drive	Mech.	Less than 1°, to many turns, depending on gear ratios	Less than 1°, to many turns, depending on gear ratios	Less than 1°, to many turns, depending on gear ratios	15–25	Yes
Load-and-fire PRIM	Mech.	Less than 1°, to many turns, depending on gear ratios	Less than 1°, to many turns, depending on gear ratios	Less than 1°, to many turns, depending on gear ratios	15–26	Yes
Multiple input	Mech.	Less than 1°, to many turns, depending on gear ratios	Less than 1°, to many turns, depending on gear ratios	Less than 1°, to many turns, depending on gear ratios	15–27	Yes

Table 16-4. Partial List of Mechanisms Used in Different Applications

Application	Mechanism
Machine Tool	Cams Clutch-brakes Genevas Star wheels Steppers
Business Machines	Cams Clutch-brakes Escapements Gears Genevas Ratchets Roll cams Steppers
Instruments	Cams Clutch-brakes Gears Genevas Inverse escapement Machine escapement Ratchets Steppers
Vending Machines	Escapements Ratchets
Production Machines	Cams Clutch-brakes Escapements Gears Genevas Ratchets Star wheels Steppers

PART II—DESIGN EXAMPLE

To conclude this study of intermittent motion mechanisms, let us consider a specific design example, a progressive case history that will show how the design of an intermittent motion mechanism is related to and affected by the design of many other portions of some complete mechanical or electromechanical system. Specifically, we will consider the design of an inexpensive electromechanical printer. Intermittent motion mechanisms will be used (a) to operate the platen that performs the actual printing operation, and (b) to advance or "feed" the paper on which the printed marks are made. Although we will concentrate on the design of the intermittent motion mechanisms in this printer, we will, of necessity, consider its other elements since they determine the input-output requirements of, and the loads on, the intermittent motion devices.

As we go along, we will relate the various steps taken, to the steps in design procedure given at the beginning of this chapter.

Printer Specifications

Our first task is to define specifications for the system we are building; in this case, the printer. These will, in turn, determine the specifications for the intermittent motion mechanisms. Let us assume that Marketing or Management (hopefully with guidance from Engineering) has established the following requirements:

> The department wants an inexpensive electromechanical printer that will record the number stored in a five-digit mechanical counter. Printout will be on command when a manual button or lever is actuated, and will be very infrequent; perhaps once a day. Since the mechanical counter will be traveling at speeds of up to 1200 cpm, however, it is desired to print and feed the paper quite rapidly whenever printout is commanded. The designer of the machine to which the printer is to be attached tells us that he can provide as much as 216 degrees of "compliance" in the counter feed. This means that the printing action could stall the counter for as much as six counts during printout. The counter would then catch up again when released. As a design target, therefore, we will try to print in approximately 0.250 second, leaving ourselves a safety factor of 50 milliseconds.

$$6 \frac{\text{counts}}{\text{printout}} \times \frac{1}{1200 \frac{\text{counts}}{\text{min}}} \times 60 \frac{\text{secs}}{\text{min}}$$

$$= 0.300 \frac{\text{secs}}{\text{printout}}$$

It is desired to use standard adding-machine-paper rolls in this printer (approximately 3-inch diameter by 3½-inch length). A sintered nylon, ink-impregnated platen is suggested to eliminate the need for ink ribbon feed mechanisms, etc., because a platen of this kind is capable of pro-

ducing many hundreds of thousands of useful impressions; ample "life" for a printer which is operated only once or twice a day.

Preliminary Definition of the Printer System

The first design step is to make some decisions regarding the locations, size, and general configuration of the counting, printing, and paper-feed mechanisms. Figure 16–1 shows a semischematic layout. The mechanical counter is mounted between two pairs of feed and idler rollers. A printing platen is located opposite the counter and is mounted on a lever. The paper roll is off to one side.

The intermittent motion drive package, shown schematically in Fig. 16–1, will have to provide mechanical outputs to operate the platen lever system and to advance the feed roll. Obviously, these two outputs will have to be sequential. We do not want to advance the paper during the actual printing operation (in some high-speed designs, of course, the paper is moved with the print wheels).

Many other printer "systems" would be possible, of course. The arrangement shown, however, would be simple and inexpensive (therefore, satisfying our basic requirements). Since we are mainly interested

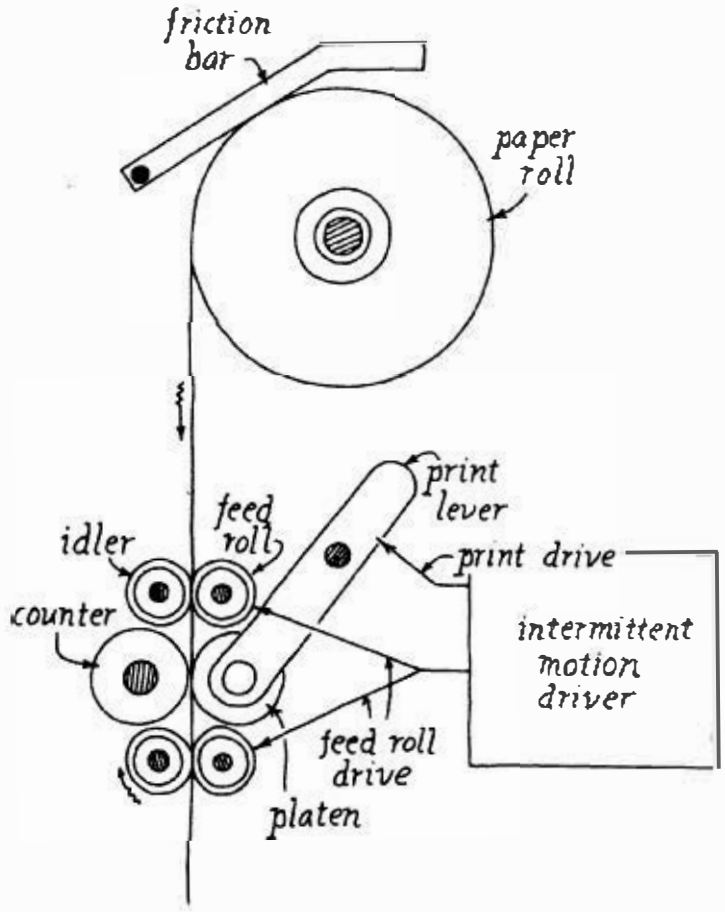


Fig. 16–1. Semi-schematic illustration of an inexpensive electro-mechanical printer: Our "design example."

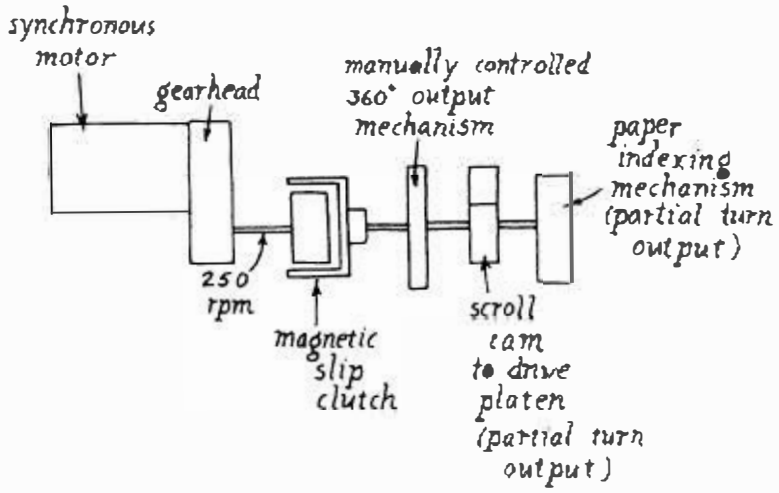


Fig. 16–2. Preliminary schematic for the intermittent motion drive system of the printer.

in illustrating the design of an intermittent motion system we need not spend too much time on refinements to the printer, *per se*.

Preliminary Definition of Intermittent Motion Drive System

The next step is to come up with a preliminary definition of the intermittent motion drive system. To repeat: two separate outputs are needed, a printing motion followed by a paper feed motion. These two motions are to be initiated by a manual control operation of some kind; someone either pushes a button or moves a control lever. The fact that the control input is manual, also means two things: First, the input will be erratic; sometimes fast, sometimes slow; sometimes complete, sometimes incomplete; sometimes smooth, sometimes hesitating; etc., depending upon the operator. Secondly, input forces will be small since a pushbutton or small lever requiring more than four pounds of actuation force is uncomfortable and difficult to use.

Figure 16-2 is a schematic of one possible intermittent motion drive system. A constant speed, synchronous, gearhead motor runs continuously, producing a steady output of 250 rpm (a little more than 4 revolutions per second or 0.240 second per revolution, slightly faster than our goal, but acceptable). This output is fed through a magnetic drag-cup slip clutch. This clutch applies a steady torque (of an, as yet, undetermined magnitude) on a first intermittent motion mechanism that is manually controlled and which will produce 360 degrees of output motion whenever it is commanded to do so. Most of the time, of course, the clutch slips and there is no output motion. The output of this 360 degree intermittent motion device drives two other mechanisms:

a cam for actuating the printing platen, and a second intermittent motion mechanism for advancing the paper feed roll. These two outputs will be sequential, but we do not yet know how much of the 360-degree output produced by the first intermittent motion mechanism should be used for the cam and how much will, therefore, be left for the paper feed. To determine this, we must go back and study the printer system in greater detail.

Notice that we first took a rough cut at the printer, then an intermittent motion system was sketched out, and now we are going back to the printer. You will find this printer-indexing mechanism–printer cycle repeating itself over and over, as we proceed. Only if you are working in a very large company and on a design effort that has been broken down by a systems engineer or someone into clearly defined isolated parts will you be designing just intermittent motion mechanisms without influencing or being influenced very much by the design of the remainder of the system.

Further Details—Printer System

It takes a considerable amount of pressure to make a successful mark in a typical printer. Ink-impregnated platens require approximately 1000 psi in a "squeeze printer." (Impact hammer printers require higher pressures because contact times are much less.) With the size and style of type planned for this particular design, 1000 psi works out to be approximately 10 lbs per wheel. The platen, therefore, will have to exert a total of 50 pounds-force on the five-wheel mechanical printer.

Since the printer is to be a relatively low-cost device we will want to use a fairly small, inexpensive motor. Judging from past experience, it is going to require a fair amount of mechanical advantage to produce 50 pounds-force on the platen with such a motor, thus, almost immediately, we go to a compound lever system as shown in Fig. 16–3. To gain additional mechanical advantage this lever is driven with a scroll cam having as gentle a slope as possible. This means we want to take up as much of the available 360 degrees as is feasible with the scroll cam, and crowd the paper feed portion of the cycle into as short an angle as possible. The height of the character on the printing wheel is 0.125 inch. A paper feed distance of 0.300 inch, therefore, would seem proper. This is approximately the length of a 60-degree arc on the feed roll. Taking this as a clue, we leave 60 degrees of motion of the intermittent

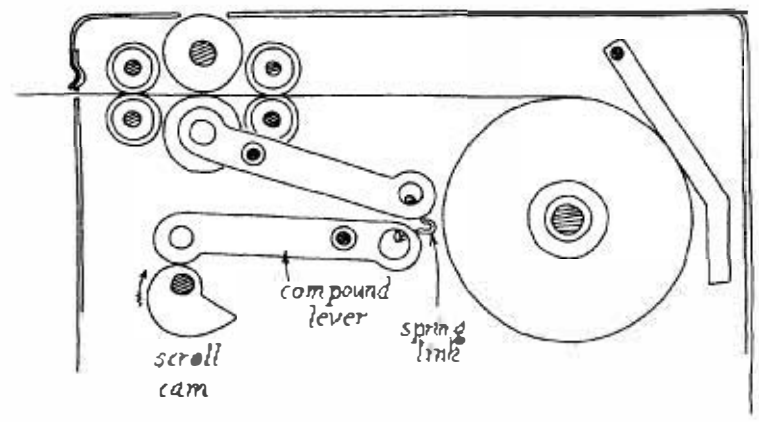


Fig. 16-3. Refinements of the printer layout, showing the compound lever system used to operate the printing platen.

motion drive assembly for paper feed and use a full 300 degrees of motion for the scroll cam.

Do not feel that there is any magic behind these various "numbers." We are groping toward a set of design specifications that will be mutually consistent, using intuition and experience (rather than "equations") to establish possible numerical values.

Detailed Specifications for the Intermittent Motion Mechanisms

Now that tentative input and output requirements for the two intermittent motion mechanisms have been defined we have arrived at Step 1 of our "Design Procedure for Intermittent Motion Mechanisms," as defined earlier in this chapter. As can be seen, a fair amount of design work has been done to arrive at the point where we can "define the primary problem" for the indexing mechanisms themselves.

In accordance with the schematic of Fig. 16-2, we have the following two indexing mechanisms to specify in detail:

A. One Turn (or 360-degree Output) Mechanism

1. Indexing rate required: 250 steps per minute
2. Indexing accuracy required: Not critical—perhaps, plus or minus 5 degrees
3. Dwell-motion pattern required: Asynchronous operation; therefore, long or short dwell periods followed by rapid indexing; output motion of 360 degrees each actuation.
4. Size of load: Medium to light
5. Cost situation: Medium to low cost
6. Secondary considerations: Manual control inputs. These will be erratic and will often occupy more time than the 0.2 second required for each output motion.

B. Paper Feed Mechanism

1. Indexing rate required: 250 steps per minute
2. Indexing accuracy required: Not critical—perhaps plus or minus 5 degrees
3. Dwell-motion pattern required: Output should dwell for 300 degrees of input motion; output should then rotate approximately 60 degrees for the next 60 degrees of input motion.
4. Size of load: Medium to light
5. Cost: Medium to low cost
6. Secondary problems: Moderate size.

Selection of Specific Intermittent Motion Mechanisms

We are now ready to select candidates for these two indexing jobs, perhaps by reference to Table 16-3. Taking the one-turn mechanisms first, list as follows, all the choices from Table 16–3 that produce 360 degrees of output motion which are usually designed to run asynchronously:

Type of Mechanism for 360° Output	Input Motion During Output Dwell	Input Motion During Output Motion	Output Motion
Simple machine escapement	0°	10° to 360°	10° to 360°
Load-and-fire escapement	<1° to many turns	Few degrees	10° to 360°
Friction escapement	Many turns	720°	360°
Variable-stroke escapement	0°	9° to 360°	9° to 360°
Spring clutch	0° to many turns	10° to 360°	10° to 360°
Low-cost electr. clutch	0° to many turns	90° to 360°	90° to 360°
Heavy-duty pawl clutch	90° to many turns	360°	360°
High performance Electr. clutch	Many turns	Many turns	Many turns
Overrunning clutches	0°	Few to 360°	Few to 360°
Roll cam	Few degrees to many turns	36° to 360°	36° to 360°
Gleason indexer	...	...	1° to 360°
Hydraulic rack actuator	Linear	Linear	30° to 360°

There are also a number of possible choices in Table 16–3, to be listed for the paper feed mechanism:

Type of Mechanism for Paper Feed	Input Motion During Output Dwell	Input Motion During Output Motion	Output Motion
Linkage ratchet	300°	60°	18°
Eccentric ratchet	320°	40°	20°
Stationary cam	324°	36°	36°
Mutilated gear	324°	36°	36°
Programmable gear	20° to 340°	340° to 20°	40° to 720°
Disc escapement	315°	45°	180°
Controlled output escapement	30°	60°	60°
Very-light-duty inverse escapement	30°	60°	22½°
6-stop external star wheel	30°	60°	60°
3-stop internal star wheel	300°	60°	120°
Roll cam	1° to Many turns	36° to 360°	36° to 360°
PRIM	1° to Many turns	1° to Many turns	1° to Many turns

The next job is to select one mechanism from each of the new listings. Let us consider the one-turn (360°) mechanism first. At least to start with, the roll cam, Gleason indexer and the various clutch systems can be eliminated on the basis of cost. The hydraulic rack actuator can be eliminated because we do not have hydraulic or pneumatic energy available. I have the feeling that a simple escapement will be adequate and will be the most economical approach, thus "machine escapement" is my choice for the one-turn mechanism.

We should note in passing that there are a number of other mechanisms listed in Table 16–3 which produce 360 degrees of output motion. These include cams, mutilated gears, external star wheels, belt drives, and the PRIM mechanisms. These have not been included in the primary list because all of them are designed to run continuously (synchronously). However, any one of them could be made asynchronous by the addition of a clutching mechanism, but I would hope to avoid this complexity in our low-cost printer. In many other design situations, however, the distinction between "normally syn-

chronous" and "normally asynchronous" would be completely unimportant.

First choice for the paper feed mechanism would be the 6-stop external star wheel, since the input and output motion characteristics are exactly what we are looking for, the mechanism is relatively inexpensive, and it does not introduce impact. Ratchets and mutilated gears would also work, but would involve rapid acceleration and/or impact. We are attempting to drive paper (with friction rolls) with the paper feed indexing mechanism and impact could result in serious slip problems. Therefore, let us select the external star wheel as a first cut.

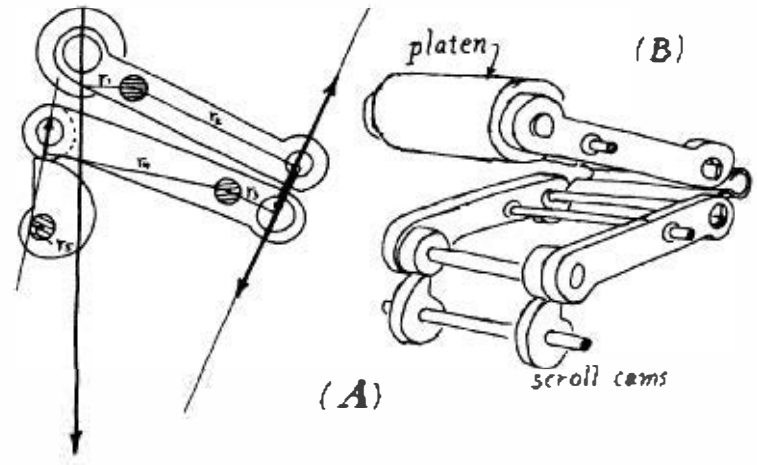


Fig. 16–5. (*Left*) Force study of platen lever system. (*Right*) Sketch of platen lever system.

Design of 360-degree Escapement

Now that candidates for the intermittent motion mechanisms have been selected, we are ready to try to incorporate these mechanisms into the design. We have, therefore, reached Step 6 in the procedure described at the beginning of this chapter.

After some preliminary sketching and several layout studies (see Fig. 16-4A, B, and C), we might arrive at the single-lever escapement shown in Fig. 16-4D. We would encounter some difficulty in designing an escapement which could be controlled manually (and, therefore, at an erratic rate) and which would still be a "true" escapement, permitting only one escape per actuation. But the mechanism shown at D, in the illustration, does this. Even if the operator should move the control (escape) lever so slowly that the scape pin nearly completes its revolution before the lever has been moved through a full cycle (which is very likely with manual control of a device producing one turn in 0.240 second), the pin on the scape wheel will strike a "safety block" preventing multiple-turn output. Either completing the stroke of the scape lever or moving it back to its original position will complete the 360-degree output motion, and trap the scape wheel. The device appears then, to be able to accommodate operator misuse or abuse.

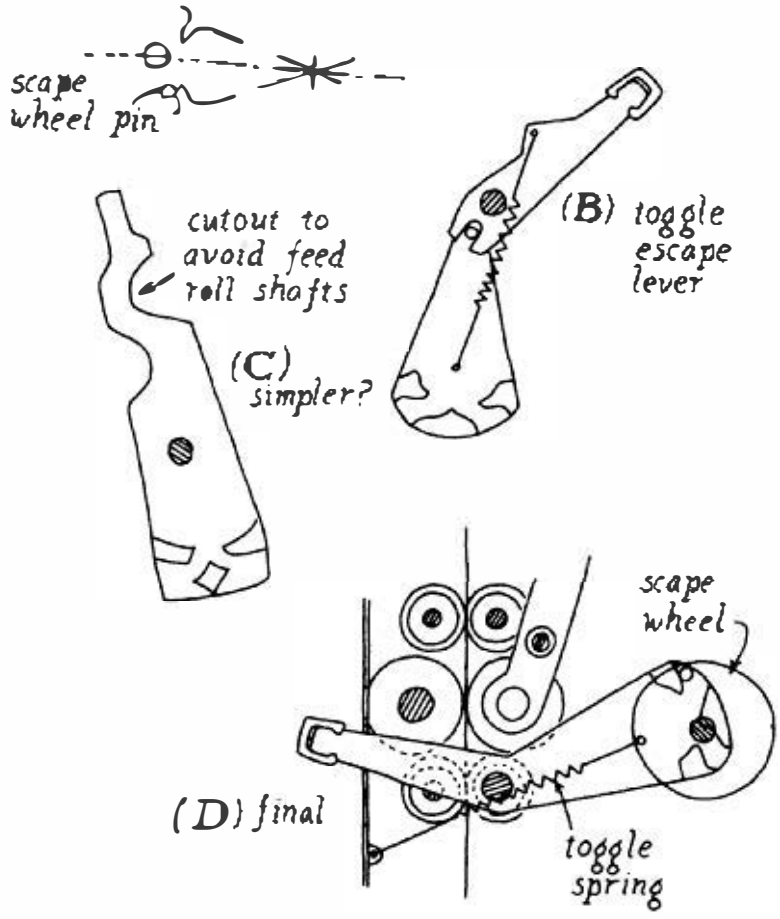


Fig. 16–4. Preliminary development (Sketch A, B, and C) of the one-turn (360°), manually operated escapement. Sketch (D) is the final version.

Six-stop Star Wheel Configuration

No specific design effort is required for the star wheel at this point in the design process, since its configuration is determined by its selection. We are, therefore, now free to move on to studies of the loads which will be placed on the two mechanisms. To do this, we must return, once again, to a consideration of the entire printer.

Load Calculations

Figure 16-5 shows the forces on the compound lever at the peak of the printing cycle. Forces are relatively light until the platen contacts the counter-

print wheel at which point they rapidly build up until 50 pounds-force is generated against the wheel assembly. This is a squeeze operation and there is no impact involved. The vector diagram (A) indicates the direction and magnitude of these forces, based on simple lever theory. In the diagram the pivots for both levers are assumed to produce little friction since it would be cheaper to provide low-cost ball or roller bearings here than to provide the heavier drive motor, which would be required to overcome bearing friction. As lever speed is small, the number of actuations required is very small (short life requirement). Factors such as preloading and precision are not required, and bearings costing only a few cents can be used.

A bearing can also be used for the cam follower. Using the methods described in Figs. 1–10 and 1–11, we can determine the resultant force exerted by the scroll cam on the first lever if we used a frictionless bearing, and compare it to the resultant produced on the same cam if we used a non-rotating follower having a maximum coefficient of friction of 0.5, for example. This comparison is shown in Fig. 16-6. As you can see, using an inexpensive ball bearing for the follower makes a significant difference in the effective ratio of the first lever.

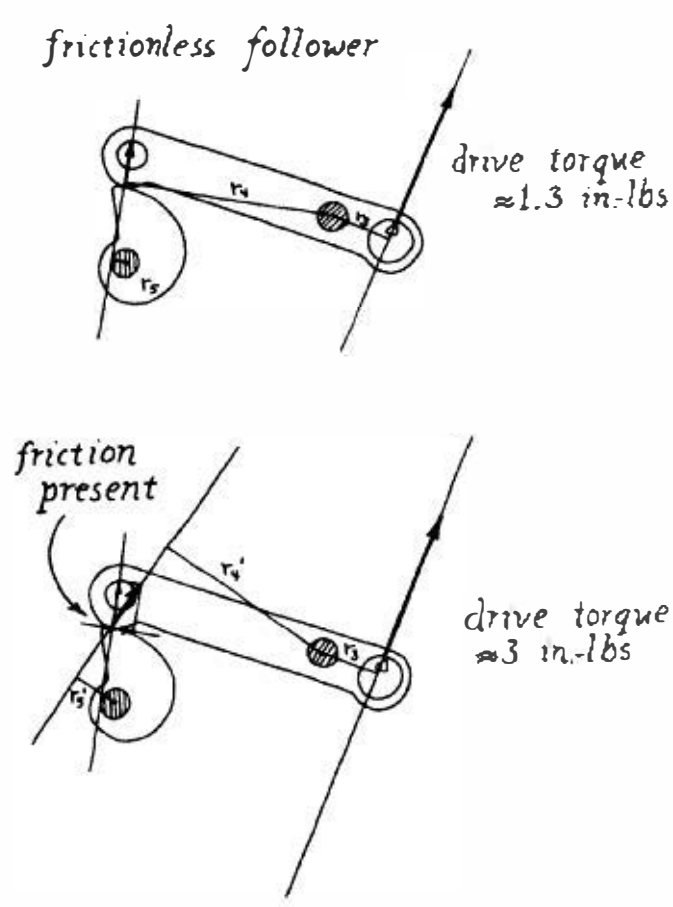


Fig. 16–6. Details of force vectors on the scroll cam follower, platen lever system, with a ball bearing follower and a non-rotating follower. Also, scroll-cam drive torque required is shown in each case.

The construction of Fig. 16–6 also reveals the scroll-cam drive torque in each case. With a plain follower the torque is about 3 in.-lbs; with a frictionless bearing it is about 1.3 in.-lbs. This is our first load calculation and we should recognize that it is only a crude approximation of the actual force. In spite of the fact that we have used frictionless bearings for the lever pivots, there will inevitably be friction in this lever system. The spring rate of the clip which connects the two levers will vary; a return spring of some sort will be required on the follower lever; dimensions will change; platen hardness will vary; etc. Nevertheless, this calculation is adequate for our present purposes.

It would be very nice if the torque required for the paper feed portion of the cycle were comparable to that required for the printing portion of the cycle—since drastic and sudden changes in torque mean rapid acceleration or deceleration—and we have seen how this can lead to vibration, impact, and general instability in most mechanisms. Our next step, therefore, will be to calculate the torque required for paper feed to see whether or not it is compatible with the printing torque.

Paper Feed Load

There are two loads involved in the paper feed. As can be seen in Fig. 16-1, a friction bar has been mounted beside the paper roll to dampen its motion and prevent the roll from coasting after it is indexed. As a first approximation we have shown a solid aluminum bar that would weigh approximately five pounds. This presumably would be replaced in a final design with a thin, spring-loaded plate. Estimating a maximum coefficient of friction of approximately 0.4, we can estimate a friction drag force of approximately 2 lbs. (See Fig. 16–7.) This force will be relatively independent of the diameter of the paper roll (although the torque exerted by the bar on the roll will, of course, vary as the diameter changes)l.

Ignoring further friction forces in the system, the only other load exerted on the intermittent motion drive during the paper-indexing portion of the cycle will be inertial loads. To determine these we must compute the inertia of the feed and idler rolls, the paper roll, and the gear train that connects the intermittent motion driver to the paper feed rolls. Since all the parts are cylindrical this is a relatively

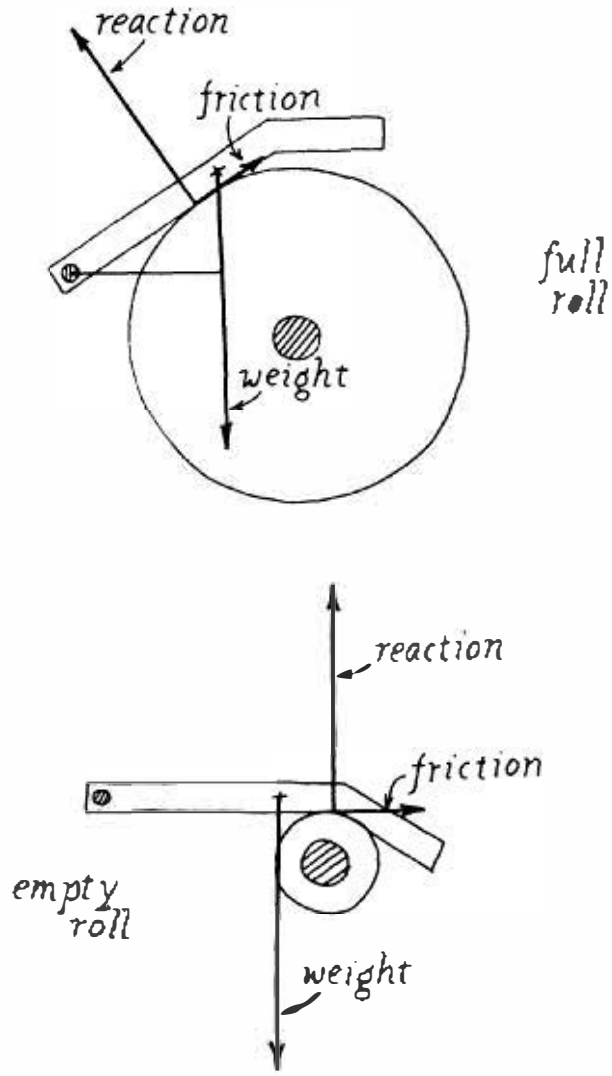


Fig. 16–7. Drag forces produced on paper roll by a friction bar.

easy task. (See appendix for inertia formulas.) Results are tabulated below:

Part	Inertia
One feed roll	0.4×10^{-4} slug-ft²
The paper roll	2.7×10^{-4} slug-ft²
One connecting gear	0.006×10^{-4} slug-ft²

There are two feed rolls and two idler rolls; all four are identical in construction, thus, the inertia given for the feed roll must be multiplied by four to get the total system inertia for these parts. There are three connecting gears, plus the output member of the star wheel whose inertia we can assume is equivalent to that of one gear; so the gear inertia must also be multiplied by four to get a rough approximation of the total inertia contributed by these members. Notice that the gears are all of equal size, thus we do not have to increase or decrease their effective inertia (or that of the feed rolls to which they are connected) by a multiplying ratio (see the section in the appendix, on inertia calculations involving gear trains, if this statement is unclear).

There is a "gear ratio" involved, however, in computing the inertia of the paper roll. The feed rollers, which are driving the paper, are $\frac{3}{4}$ inch in diameter and rotate 60 degrees with each step. The paper roll is approximately 3 inches in diameter and is pulled by the paper. The full paper roll will, therefore, turn only $60° \div N$; N being the ratio of the diameter of the paper roll to the diameter of the feed roll. And the inertia of the full paper roll as it affects the drive system will be $1/N^2$, or 1/16 times the inertia of the paper roll about its own axis. This means that the reflected inertia of the paper roll on the feed roll and intermittent motion mechanism drive system is:

$$\frac{2.7 \times 10^{-4}}{16} = 0.17 \times 10^{-4} \text{ slug-ft}^2$$

Combining all these inertias, we get a total system inertia of:

$$\begin{aligned} &1.60 \times 10^{-4} \text{ slug-ft}^2 \\ &0.17 \times 10^{-4} \text{ slug-ft}^2 \\ &0.024 \times 10^{-4} \text{ slug-ft}^2 \\ \hline &1.794 \times 10^{-4} \text{ slug-ft}^2 \end{aligned}$$

Since the calculations are rough, and ignore some friction factors, etc., let us call this 2×10^{-4} slug-ft² to be safe.

Determining inertia of the feed system does not, of course, determine the torque required to index it. To do this we must determine the acceleration of this inertial system and then find torque by using the familiar equation:

$$\tau = I\alpha$$

Motion Curves

To determine the acceleration of the feed system we must construct motion curves for the mechanism which is producing this motion, namely, the six-stop star wheel. We start by making a layout of the star wheel, then some paper or plastic cutouts of input and output mechanisms. Next, assuming constant input speed (which we would have with a synchronous motor drive as long as we do not exceed the capacity of the slip clutch), we construct a displace-

ment-versus-time curve for the star wheel. Such a curve is shown in Fig. 16-8. Notice that we have used input displacement rather than time for the horizontal axis. As explained in Fig. 2-9, this is frequently more convenient in layout work, and we can convert displacement to time later on if the input is running at constant velocity, as it is here. A simple calculation shows that 60 degrees of input is equivalent to 40 milliseconds when the input speed is 250 rpm. We also must calculate the output displacement (60 degrees) in radians: 60 degrees equals 1.05 radians.

We now use graphical differentiation to derive velocity and acceleration curves for this mechanism.

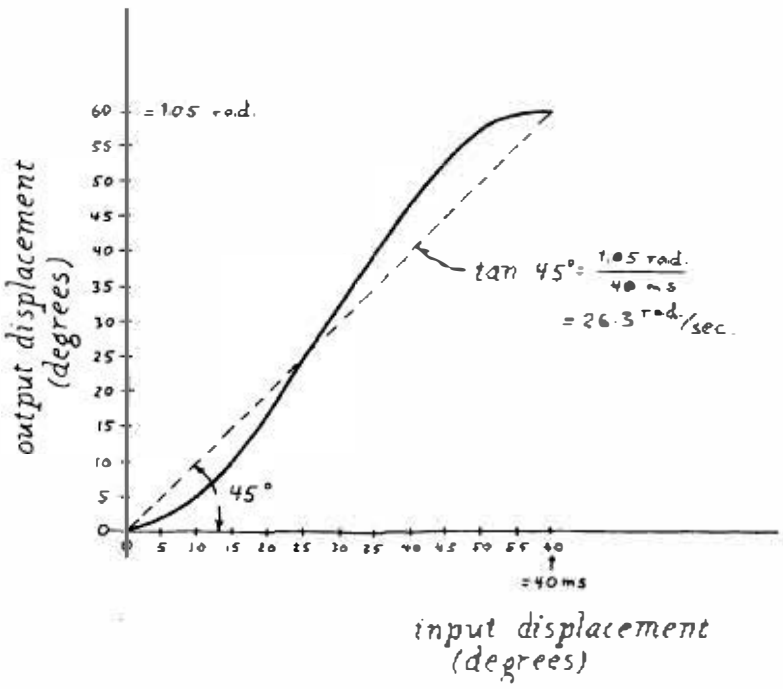


Fig. 16-8. Output displacement versus input displacement for the six-stop star wheel. The construction to determine the vertical scale of the velocity curve is also shown.

These are shown in Figs. 16-9 and 16-10. As in Chapter 2, I must emphasize that curves obtained in this manner are approximations only. Nevertheless, they are very useful for roughly estimating the magnitude of peak velocities and accelerations in a given system.

The vertical axes for the velocity and acceleration curves are given in terms of tangent or slope. We are interested in knowing the peak acceleration of the system in terms of radians/sec², and so, must convert these slope values to the proper units. To get the correct velocity scale we determine the velocity represented by a tangent of 1.0 (slope angle of 45 degrees) on the displacement graph. This equals:

$$1.05 \div 40 \times 10^{-3} = 26.3 \text{ radians/second}$$

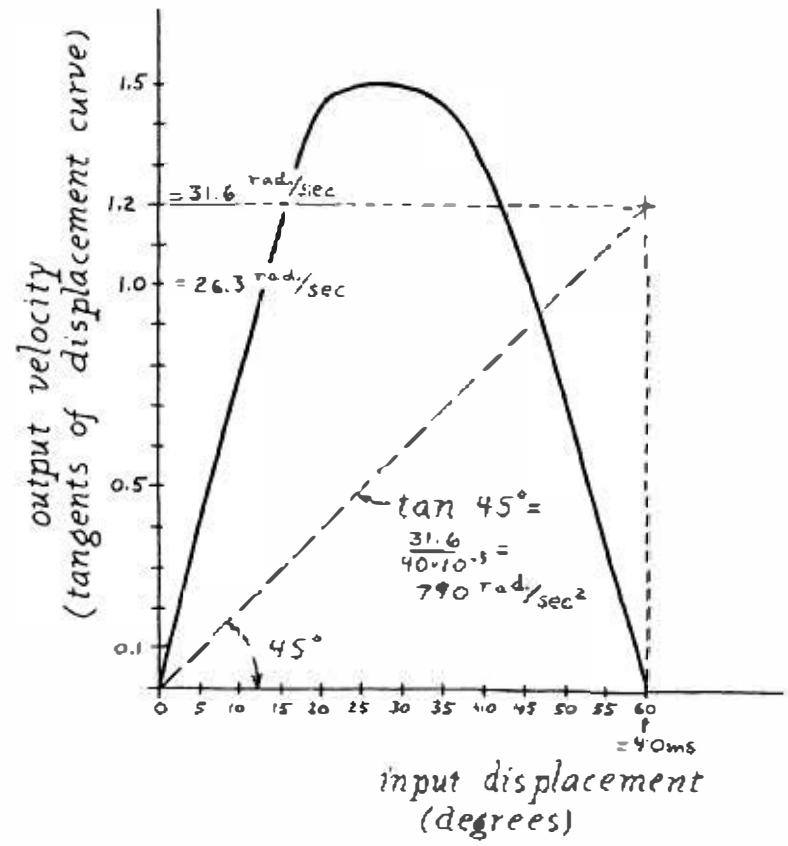


Fig. 16-9. Output velocity versus input displacement for the six-stop star wheel. The construction to determine the vertical scale of the acceleration curve is also shown.

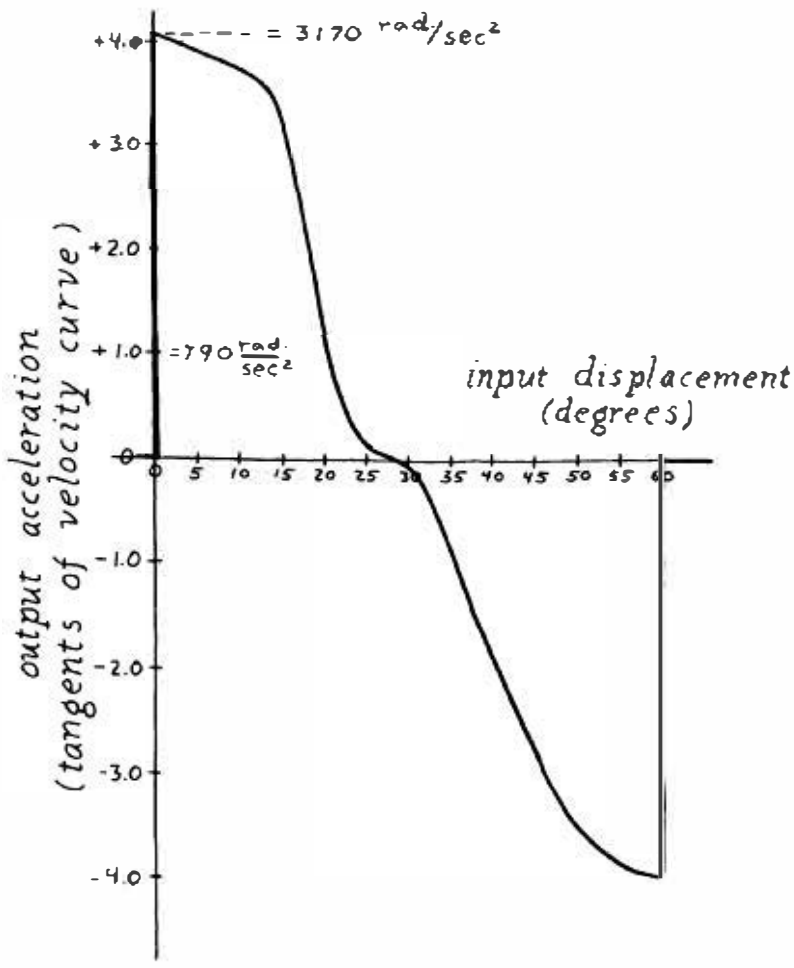


Fig. 16-10. Output acceleration versus input displacement for the six-stop star wheel.

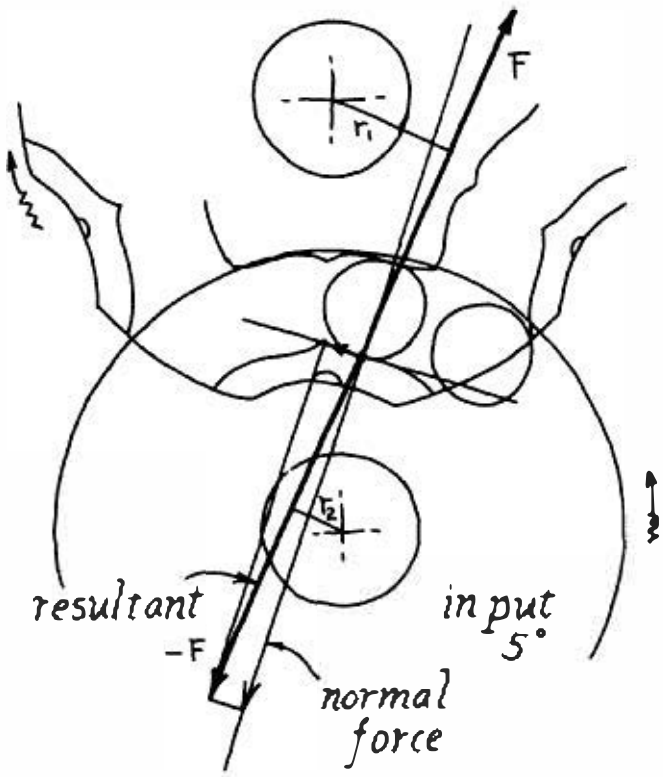


Fig. 16–11. Analysis of the input torque required by the star wheel after the driver has rotated five degrees from start of motion cycle. The geometry of the drive determines the direction of the forces on both members (we have to make an assumption regarding the coefficient of friction). This determines the "lever arms" for input and output. Our previous output torque calculations now allow us to determine the magnitude of the forces and hence, input torque. (See Fig. 1–12.)

This tells us that "1.0" on the vertical axis of the velocity curve represents a velocity of 26.3 radians/sec. On this same scale 1.2 units would represent a velocity of 1.2/1 × 26.3, or 31.6 rad/sec.

Making a similar calculation now on the velocity curve (determining what a 45-degree slope is equal to, in acceleration terms) we find that:

$$\text{Tangent of } 45° = 31.6 \div 40 \times 10^{-3}$$
$$= 790 \text{ radians/second}^2$$

So "1" on the vertical axis of the acceleration curve represents an acceleration of 790 radians/second². The peak slope on the acceleration curve is 4.01 in "slope units." This must correspond to:

$$\frac{4.01}{1} \times 790 = 3{,}170 \text{ radians/second}^2$$

The accelerations at other input angles (or times) can now be determined in a similar fashion.

We can now multiply the accelerations at points of interest by the inertia of the system to determine the total torque required to overcome the inertia of the paper feed system For example:

$$\text{Max. torque} = I\alpha$$
$$= 2 \times 10^{-4} \times 3170$$
$$= 6.34 \times 10^{-1} \text{ ft-lbs}$$
$$= 7.6 \text{ in-lbs}$$

In addition, we must supply torque to overcome the frictional drag produced by the friction bar on the paper roll. We estimated that this produced a 3-lb drag force, which would be transmitted through the paper to the $\frac{3}{16}$-inch radius feed roll where it would produce a torque of $\frac{3}{16} \times 3 = 0.563$ in-lbs. (Alternately, we could calculate the drag torque on the paper roll and relate this to input torque required, by applying gear ratio N, again.)

Total peak drive torque required, then, is approximately 7.6 + 0.563, or 8.2 in-lbs.

At first glance, then, we apparently need almost three times as much drive torque to index the paper roll and the associated paper feed system, as would be needed to operate the platen during the printing cycle; but this is not true. What we have just calculated is the torque we must produce in the *output* of the six-stop external star wheel to index the system. We must now calculate the torque required from the star wheel input to produce this amount of

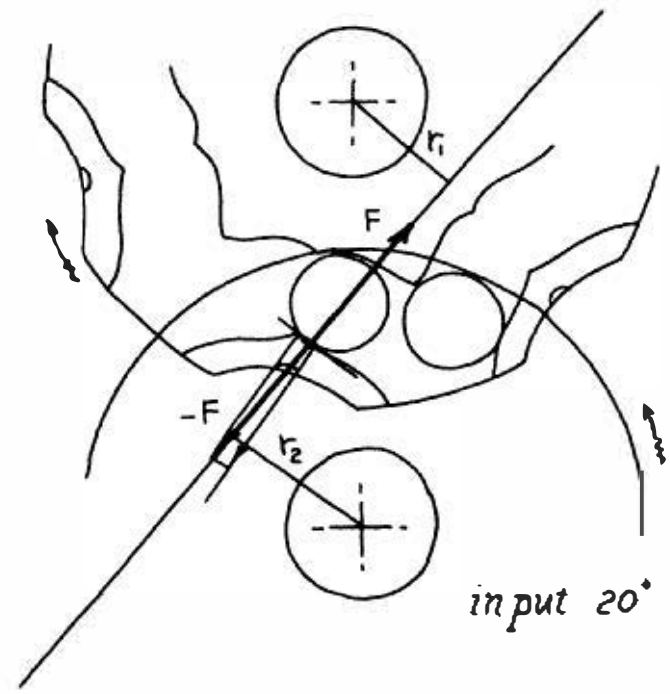


Fig. 16–12. Analysis of input torque requirements of the star wheel after the driver has completed twenty degrees of the input motion portion of the cycle.

output torque; again using the type of vector analysis shown in Figs. 1–10 or 1–12. When making such an analysis it is best to study several different positions of the mechanism to be sure that you have not overlooked the peak torque point.

Figures 16–11, 12, and 13 show three torque constructions for the star wheel. We have assumed a coefficient of friction of only 0.1, since we assume the drive pins in this star wheel are miniature rollers, and experience tells us the friction in such a system is low. Notice that Figs. 16–11 and 16–12 represent various positions of the star wheel during the acceleration phase of the indexing motion, while Fig. 16–13 represents the end of the deceleration phase. Notice also that the peak torque required of the input member is considerably less than that produced in the output member. At first glance this seems impossible, until one recognizes that the peak torque in the output is generated at a time when the output is moving very slowly, at nearly zero velocity, in fact. (At the very beginning and the end of the indexing cycle.) The velocity ratio between input and output is at a maximum, therefore, when output torque requirements are greatest, and we have, in effect, a large "gear ratio" when we need it the most. When input and output of the star wheel are moving at essentially the same velocity (in the middle of the indexing cycle) output acceleration is 0, or near 0, and drive torque requirements are correspondingly low.

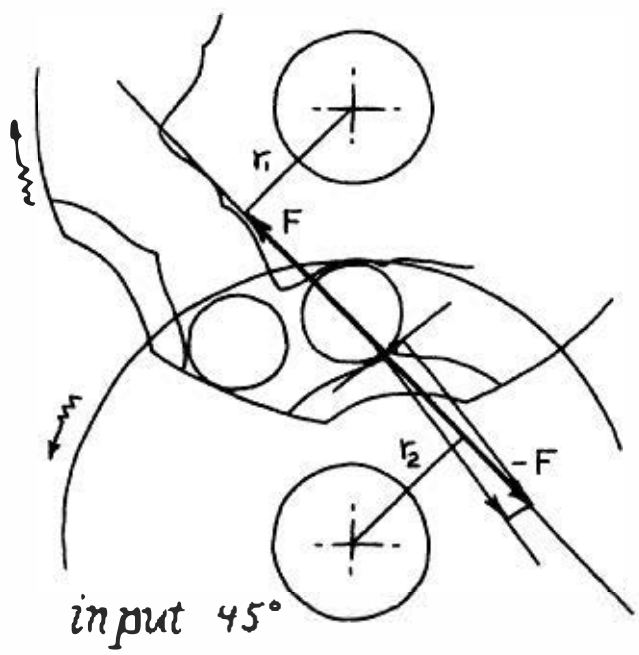


Fig. 16–13. Analysis of drive torque requirements of the six-stop star wheel after the driver has completed 45 degrees of the input portion of the cycle. Note that we are now dealing with a negative torque (deceleration).

As a result of a complete analysis of the sort described and illustrated above, we would discover that the peak drive torque required is approximately 3 in.-lbs, which is identical to the torque required to operate the printing platen when we used a non-rotating follower (Fig. 16-6). Such a follower, therefore, should probably be used to smooth drive torque requirements. Using a ball bearing follower would result in a sudden torque increase after 300 degrees of input motion.

Reconsideration of Intermittent Motion System

We now have a fairly good idea of the requirements of the intermittent motion drive system. In addition to the original specifications, we now know what loads will be imposed on the driver. And this raises several problems: our calculations indicate we need approximately 3 in-lbs to operate the platen or the feed mechanism. To be safe, we should probably provide a slip clutch with 4 in-lb capability. This is a lot of torque for a small magnetic-drag-cup slip clutch. We will probably have to use a friction-disc slip clutch. Another problem: 4 in-lb. of slip continuously at 250 rpm works out to be approximately 16 watts of power dissipation. The original intention, to let the motor run continuously and the clutch slip continuously, despite printing only once a day, looks pretty silly if this much power is involved. We should probably turn off the drive motor in-between printout commands, reducing power loss and clutch wear considerably.

At this point the original indexing system concept could be abandoned altogether, and perhaps we could look for a clutch-brake system as a replacement for the entire synchronous motorslip clutch-escapement train. (Step 7 in our Design Procedure.) I would still be concerned about the cost of the clutch-brake approach, however, and suggest instead, that we place a small flap or cover over the button or lever used to actuate the escapement. The operator must lift this flap to get at the escapement control, and the act of lifting can be used to actuate a switch to start the motor. An on-off switch would have to be provided for this printer in any event, so this design modification does not add significantly to the cost of the system. By actuating the switch with a flap of this kind, we give the motor time to start and attain full speed before the operator can reach and operate the escapement. Only a fraction of a second will be required, but this is too much time to try to build it into the 0.240 second available

for the print cycle; we must get the motor started first.

The high torque required by the system causes one additional concern. The original escapement design (Fig. 16-4) shows a small-diameter pin being used as the scape tooth on the scape wheel. I would be very concerned about the life of such a pin under a 4 in-lb load. The pin, in fact, is located at about 0.6-inch radius on the wheel, and would, therefore, see a static force approaching 7 lbs each time it brings the system to a stop. Furthermore, since some inertia is involved, the 7 lbs would be increased by impact. Also, the escapement should be redesigned to provide a much sturdier "tooth." Judging from the difficulties experienced with a 360-degree escapement (whose geometry dictated a small pin), we

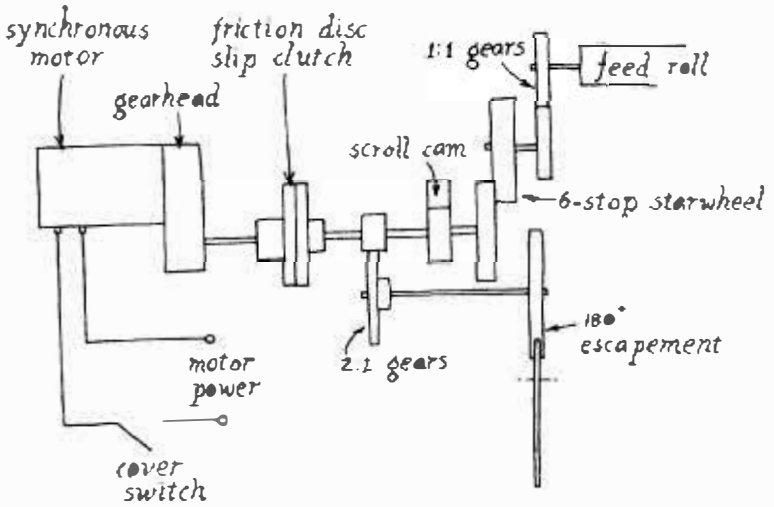


Fig. 16-14. Revised schematic of the intermittent motion drive system showing the additions of a cover switch for the motor and a two-to-one gear ratio between the control escapement and the rest of the system.

should, instead, probably try for a 180-degree escapement, geared up two-to-one to the rest of the system. We then will be able to use a conventional escape tooth rather than a pin.

A schematic of the revised indexing system accommodating the new type of slip clutch, the cover switch for the motor, and the 180-degree escapement is shown in Fig. 16-14.

Notice again, that these changes in the indexing system have been dictated by the requirements of the total system design. Perhaps the entire printer could be redesigned to reduce drive torque requirements and make it possible for us to use the original indexing system; however, this would be using the ship to steer the wheel. In most design situations we will probably do what has been done here—modify the indexer to accommodate the machine.

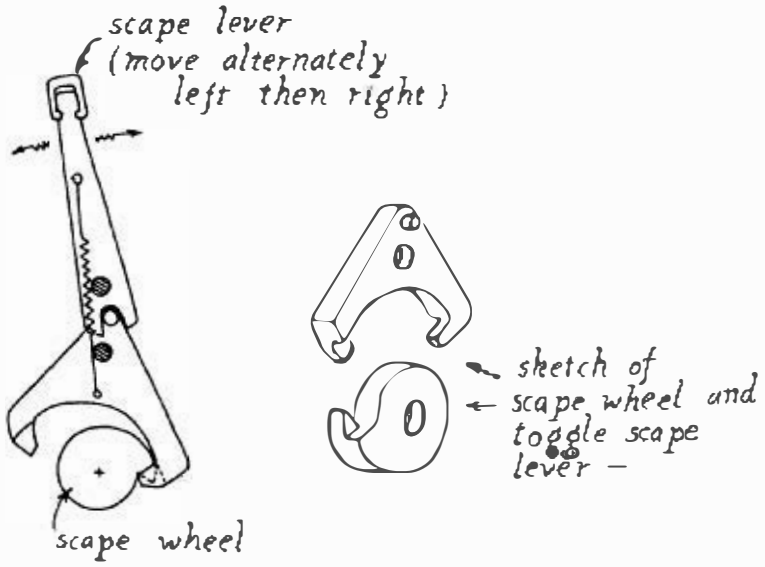


Fig. 16-15. One possibility for a 180-degree output, manually operated escapement for the printer.

Redesign the Escapement

Figures 16-15 and 16-16 show two new escapement designs, each producing 180 degrees of output motion, each very tolerant of variations in the speed and duration of input control stroke, and each involving scape wheel and lever teeth of significant size and, therefore, of significant strength. Note that in going from a 360-degree output to 180 degrees we have, of necessity, doubled the torque which must be controlled by the escapement; this is an undesirable change. But the gain in scape-wheel tooth

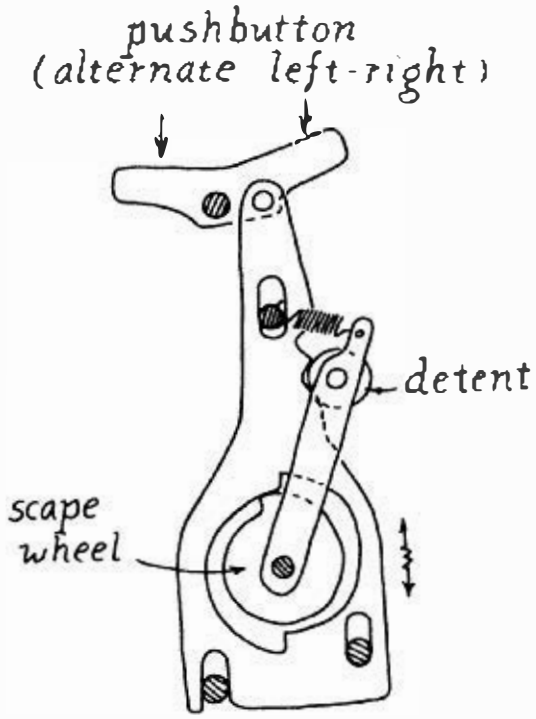


Fig. 16-16. A second 180-degree output, manually operated escapement. Either this design or that of Fig. 16-15 would probably be acceptable for the printer design.

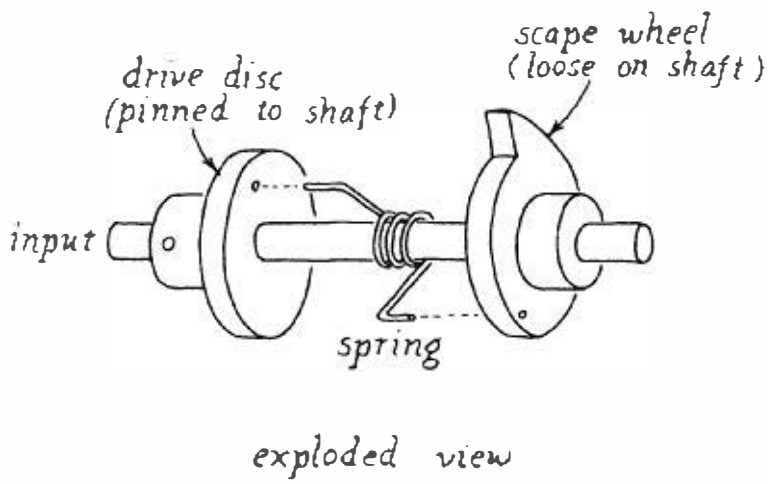


Fig. 16–17. Showing a possible method of providing some compliance between the scape wheel and the rest of the system, to reduce impacts in the escapement.

strength is even greater than the increase in torque and thus, more than compensates for this change. It is something to consider a little later, however.

Both escapements shown in the illustrations are "deadbeat" escapements; the scape wheel does not have to back up when the scape lever is actuated. In the case of Fig. 16–15 this is achieved by putting a proper radius on the scape wheel; in the case of Fig. 16–16 we achieve it by using a reciprocating scape lever which is extracted radially.

Note that in Fig. 16–15 the requirement for a deadbeat escapement necessitates the use of a two-level scape wheel tooth. This requirement somewhat complicates the shape of the scape wheel and scape lever, although these pieces could still readily be made of powdered metal, or could be cast or machined. Personally, I am inclined to favor the mechanism of Fig. 16–16. Either one, however, would probably be satisfactory.

As mentioned before, the fact that the torque controlled by the escapement has been doubled is some cause for uneasiness. In order to reduce impacts on this part, it might be desirable to place some compliance between the scape wheel and the slip clutch, as suggested by Fig. 16–17. The scape wheel teeth would still control the same static force, but the force would be applied gradually, impact effects would be eliminated, and the life of the mechanism would undoubtedly be extended. The cycle time would not be increased.

One further consideration relating to the escapement must be the amount of force required to actuate it. Assuming a coefficient of friction of approximately 0.5 (which is on the high side and could presumably be reduced by periodic lubrication of the escapement), and assuming that we are using the mechanism shown in Fig. 16–16, I estimate the required control force to be on the order of magnitude of two to three lbs. This seemed on the high side when I first calculated it, so I measured the force required to actuate several different "buttons" on commercial equipment in my employer's laboratory. I found that these ranged from one, to a little over four lbs. The one-lb button was so light that it was unsatisfactory. There was no feeling of having accomplished anything when it was pushed. At the other end of the scale, the four-lb button, although still manageable, was a little uncomfortable. I concluded, therefore, that a two- to three-lb escapement control force would be more acceptable.

Final Details

There are many other factors which could be considered in the complete design of a printer, of course.

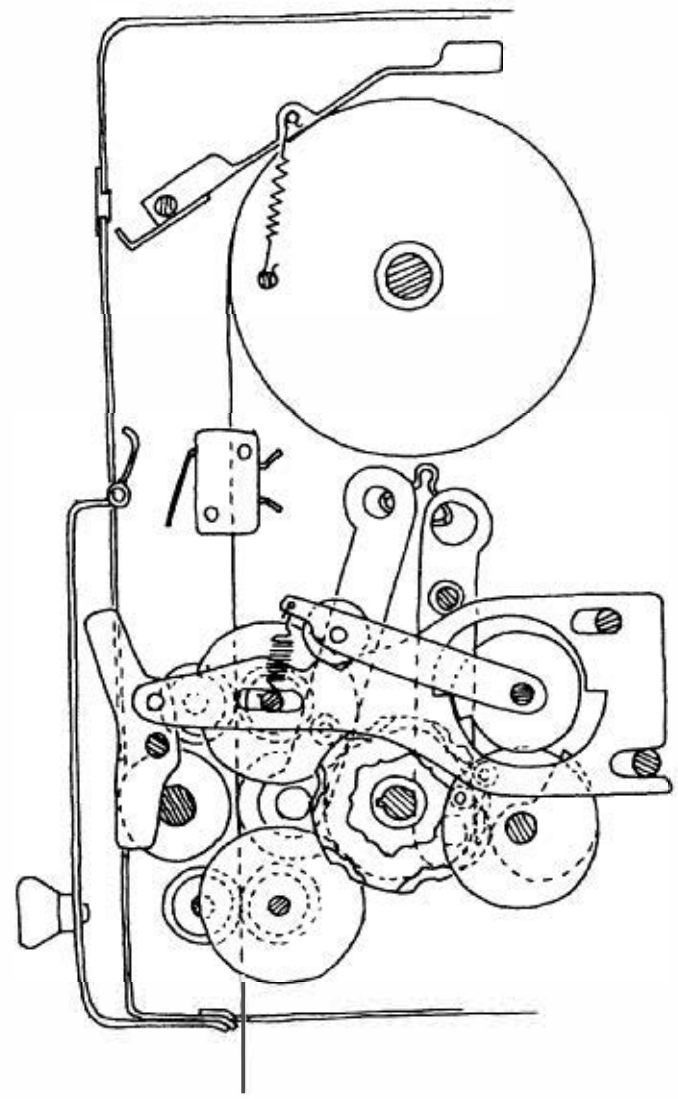


Fig. 16–18. Final layout of the intermittent motion drive system showing its relationship to principal elements of the printer.

The feed rolls should be crowned slightly to prevent the paper from walking toward one side. Edge guides should, perhaps, be provided at critical points for the paper. The counter should be mounted near the cover so that the operator can read printed numerals on one side while embossed characters on the other side are printing. An indicator of some kind should be provided to warn the operator when the paper is nearing depletion. The cover of the printer should be hinged in such a way that the feed rolls and paper roll can be reached easily for replacement of the paper. The idler rolls should be mounted on small levers so that they will be able to cramp and grip the feed rolls.

The drive motor should be capable of producing a little more torque than the calculations indicate is required in order that there will be a safety factor to offset approximations and errors in the calculations. The motor tentatively selected for this design would produce approximately 5.8 in-lbs at 250 rpm. This motor is 4½ inches in diameter and 5½ inches long; a little larger than I would have liked, but still usable.

As mentioned earlier, it would probably be more desirable to provide an adjustable spring-loaded drag plate to control the paper roll rather than the dead-weight shown in the earlier illustrations. The drag torque would then be adjustable, another safety factor. The slip torque of the slip clutch could also be made adjustable.

There are many other factors, I am sure, which should and would be considered in the design of an actual printer, but I hope this one example has served to illustrate the type of procedure one must follow in designing an actual intermittent motion mechanism for a particular design situation. Again, it is most important to realize the degree of interplay that exists between the design of the indexer and the design of the total system. Each influences and is influenced by the other, and it is only when the entire system has been brought into balance that the designer can consider his job finished.

Figure 16-18 shows a partial layout for the final printer design. Many elements are still lacking, but the principal elements of the intermittent motion drive system are all present and accounted for.

Appendix

I. TABLE OF SYMBOLS AND SIGNS

Symbols

The following symbols are used in this text (see Table, Appendix II for units not listed below).

A = Area
a = Linear acceleration
AC = Alternating current
B,b = Base of triangle used to measure slope of curve
C = Capacitance (microfarads)
C = Damping constant (lb-sec/ft)
Cc = Critical damping constant (lb-sec/ft)
cg = Center of gravity
cp = Center of percussion
D = Linear displacement
DC = Direct current
E = Voltage (volts)
F = Force
g = Acceleration due to gravity
H,h = Height of triangle used to measure slope of curve
I = Moment of inertia (slug-ft^2)
I = Electrical current (amperes)
J = Jerk (rate-of-change of acceleration—ft/sec^2/sec)
K = Spring constant (pounds/ft)
K = Coefficient of restitution (dimensionless)
L = Length (ft)
L = Inductance (henries)
m = Mass (slugs)
N = Gear ratio
Q = Energy lost during impact (ft-lbs)
R,r = Radius (ft)
R = Electrical resistance (ohms)
t = Time
Δt = Elapsed time
v = Linear velocity
v' = Linear velocity following a collision or impact between two or more bodies (ft/sec)
W = Weight (lbs)
X,Y,Z = Coordinate axes
α = Angular acceleration
ρ = Density (slugs/ft^3)
β = Angle (radians)
θ = Angle (radians)
θ = Angular displacement (radians)
τ = Torque (ft-lbs)
ω = Angular velocity (radians/sec)
ω' = Angular velocity following a collision or impact between two or more bodies (radians/sec)
∞ = Infinity
● = Grounded pivot or bearing
○ = Moving (translating) pin or bearing
= Signifies motion in the direction indicated
= Differential gear

Sign Conventions

Linear vector quantities (linear force, acceleration, velocity, displacement, etc.), are considered positive if upward, or to the right, in the XY plane. They are considered negative if downward, or to the left.

Rotational vector quantities (angular torque, acceleration, velocity, displacement, etc.) are considered positive if operating in a clockwise direction (looking toward the body along the axis of rotation or torque). They are considered negative if counterclockwise.

II. TABLE OF COMPATIBLE UNITS*

(Stick to any one column and you will stay out of trouble.)

Quantity	English		CGS	SI
Force	lb	ounce	dyne	newton
Acceleration	ft/sec^2	in./sec^2	cm/sec^2	m/sec^2
Velocity	ft/sec	in./sec	cm/sec	m/sec
Displacement	foot	inch	cm	meter
Mass	slug	oz sec^2/in.	gram	kilogram
Torque	ft-lb	in.-oz	cm-dyne	newton-meter
Angular Acceleration	rad/sec^2	rad/sec^2	rad/sec^2	rad/sec^2
Angular Velocity	rad/sec	rad/sec	rad/sec	rad/sec
Angular Displacement	radian	radian	radian	radian
Moment of Inertia	slug-ft^2	inch-ounce-sec^2	g-cm^2	kg-m^2
Mass Density	slugs/ft^3	oz/in.^{3}	g/cm^3	kg/m^3
Volume	ft^3	in.^{3}	cm^3	m^3
Acceleration due to Gravity	32 ft/sec^2	384 in./sec^2	980 cm/sec^2	9.8 m/sec^2
Work	ft-lb	in.-oz	dyne-cm (erg)	joule
Potential Energy	ft-lb	in.-oz	dyne-cm (erg)	joule
Kinetic Energy	ft-lb	in.-oz	dyne-cm (erg)	joule
Impulse	lb-sec	oz-sec	dyne-sec	newton-sec
Momentum	slug ft/sec	oz in./sec	g cm/sec	kg m/sec
Time	second	second	second	second
Voltage	volt	volt	volt	volt
Current	ampere	ampere	ampere	ampere
Spring Constant	lbs/ft	ozs/in.	dynes/cm	newtons/meter

* If engineers were as rigorous as physicists they would be more familiar with the "correct" units of force in the SI and C-G-S systems of units: these correct units being the newton and dyne respectively. Unfortunately, we use spring scales and balance weights that are calibrated in grams and kilograms (units of mass) on the one hand, or calibrated in pounds and ounces (units of force) on the other. I strongly recommend that the reader convert data from "gram of force" or "kilogram of force" units to dynes and newtons whenever he encounters them. You can use the following formulae for converting:

$$\frac{\text{Grams of force}}{980} = \text{dynes}$$

$$\frac{\text{Kilograms of force}}{9.8} = \text{newtons}$$

III. GRAPHICAL INTEGRATION

There are several ways to integrate a curve graphically. Integration, of course, merely consists of "finding the area under" the curve. One way to do it is to draw the original curve on a piece of graph paper and then count the squares under the curve to determine area. The "value" of each square is determined by the scale factors used to plot the original curve, as explained in Chapter 1. (See Fig. A–1.)

Another method is to use algebraic equations to calculate areas under larger portions of the curve. This, of course, is only possible if these areas can be broken down into simple shapes such as rectangles, triangles, squares, etc. (See Fig. A–2.)

A third method is to use the instrument shown in Fig. A–3, which is called a *planimeter*. A stylus on one arm of the instrument is moved along the curve and along the axis which bounds the area to be measured. A small counting device on the planimeter then gives the total enclosed area. Again, the units of area must be related to the scale factors of the original curve.

IV. GRAPHICAL DIFFERENTIATION

There are several ways to find the differential of a curve at a given point, graphically. The first step in each case is to draw a line which is tangent to the curve at the point where the slope of the curve is to

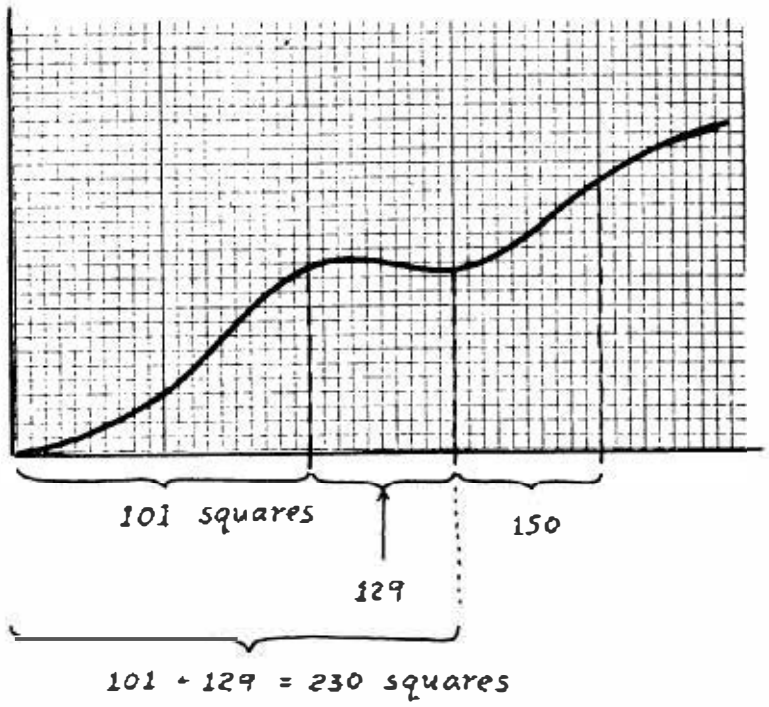
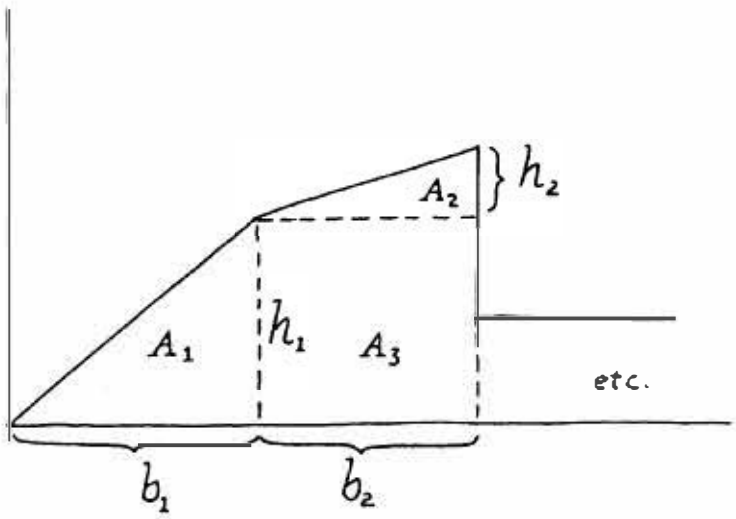


Fig. A-1. Counting the squares under a curve to integrate it graphically.



$$A_1 = \frac{1}{2} b_1 h_1$$
$$A_2 = \frac{1}{2} b_2 h_2$$
$$A_3 = b_2 h_1$$

Fig. A-2. Algebraic formulas can be used to plot the area under a curve.

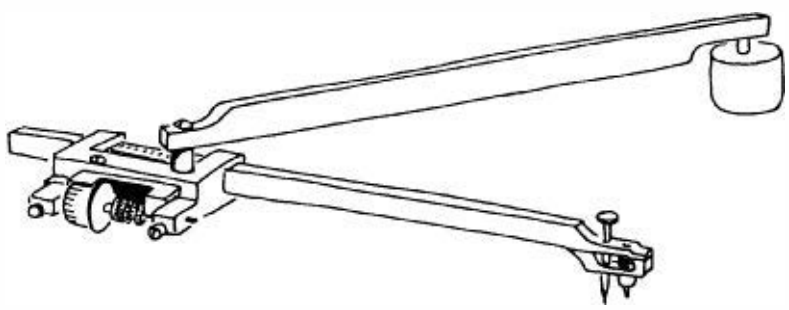


Fig. A-3. The planimeter, an instrument used to compute areas, can be used for graphical integration.

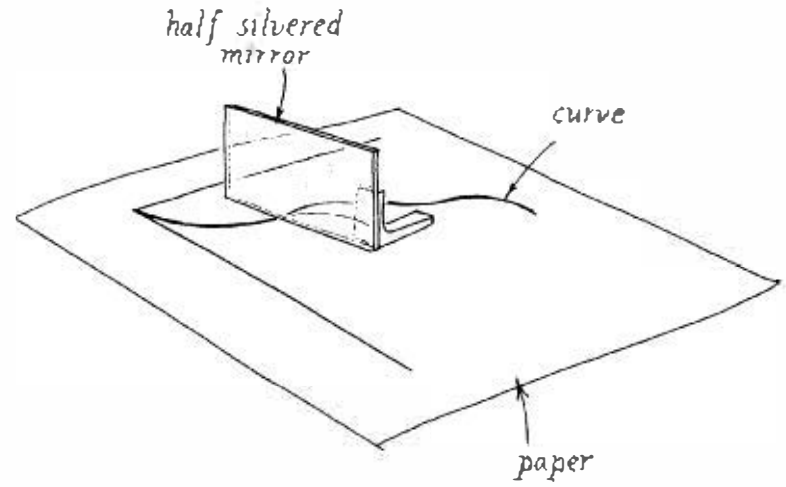


Fig. A-4. A half-silvered mirror, held perpendicular to the plane of the paper, will help you draw a line normal (perpendicular) to the curve. Draw a line along the bottom of the mirror after turning it so that the two images seen in (and through) the mirror coincide.

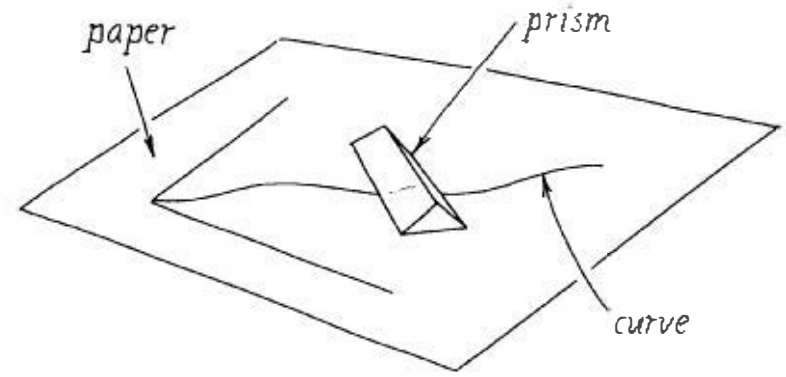


Fig. A-5. A prism can also be used to construct normal lines. The slope or tangent line, of course, is drawn at right angles to the normal line.

$$\text{tangent } \theta = \frac{1.60}{1.70}$$

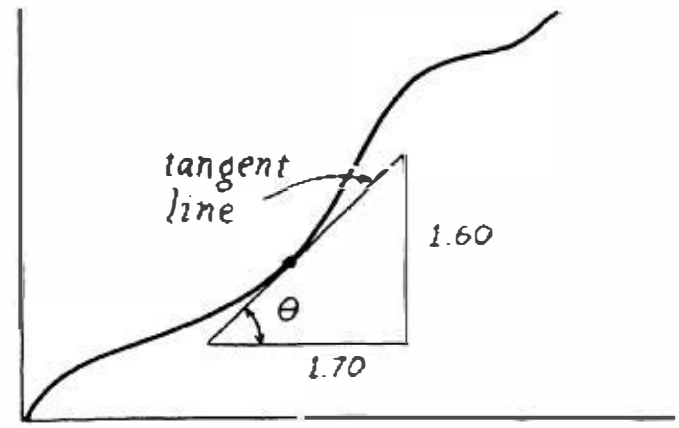


Fig. A-6. Slopes can be calculated algebraically. ("Slope" equals the tangent of the angle θ.)

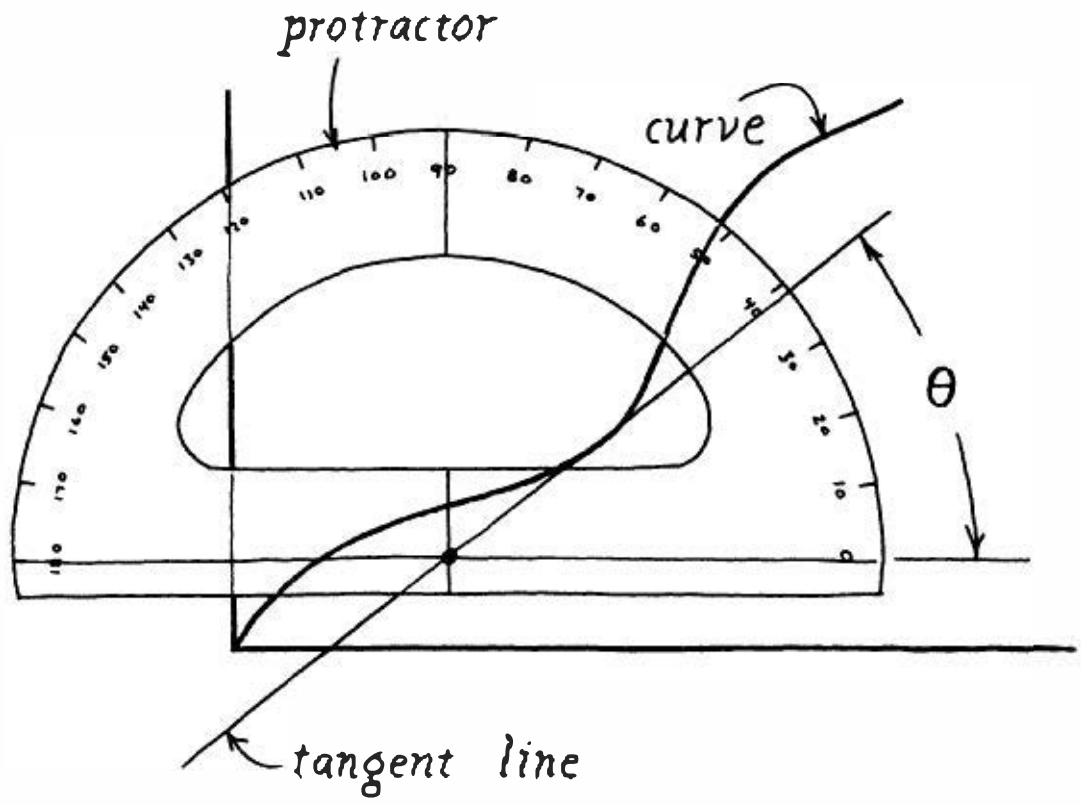


Fig. A-7. A protractor can be used to measure angle θ, whose tangent is then found in a trigonometric table.

be determined. There are three ways to construct such tangent lines which will be described here.

In many cases, you can merely draw the line in "by eye." If the curve is geometrically simple, and consists of a series of triangles, rectangles, or gentle curves, etc., then, of course, this is a very easy thing to do. If the curve is complex, the tangents you draw by eye may be in error to some extent, but this may not affect your final results too drastically if all you are after is an approximate "solution."

There are two instruments which can be used to improve the accuracy of the tangent line. One of them consists of a half-silvered mirror held perpendicular to the plane of the paper on which the original curve is drawn. (See Fig. A-4.) When the operator looks in this mirror he can see two lines. One of them is a reflection of the curve in front of the mirror, and dimly seen is the other, the remainder of the curve in back of (through) the mirror. The mirror is now rotated until these two images come together and appear to describe a single curve. A line is then drawn along the bottom of the mirror. This line is normal (or perpendicular) to the curve at the point of intersection. A drafting machine or a pair of triangles can now be used to draw a tangent line at this point, the tangent line of course, being perpendicular to the normal line.

A prism can also be used to find a normal to a curve. The prism is placed over the curve and the draftsman looks down into the prism at the image of the line which appears within it. In most orientations, the curve will appear to have been broken by the prism, which is now rotated until the curve once more appears to be a continuous line. A line drawn along the edge of the prism is again normal to the curve in this region and it can be used to construct a tangent line. (See Fig. A-5.) The prism technique, I think, is a little less accurate than the mirror technique where there arc rapid changes in the curvature of the line being differentiated, but both have their place.

Once the tangent line has been drawn, we can proceed to determine the slope of that line; the slope being the tangent of the angle formed by that line and the horizontal axis. There are several ways to determine this slope.

The slope can be determined trigonometrically as shown in Fig. A-6 where we measure the height and base of the triangle formed by the tangent line and then divide one by the other to determine the tangent of the angle. Height and base, of course, must be measured in the units or scale factors used to plot the original curves.

Another technique is to use a conventional pro-

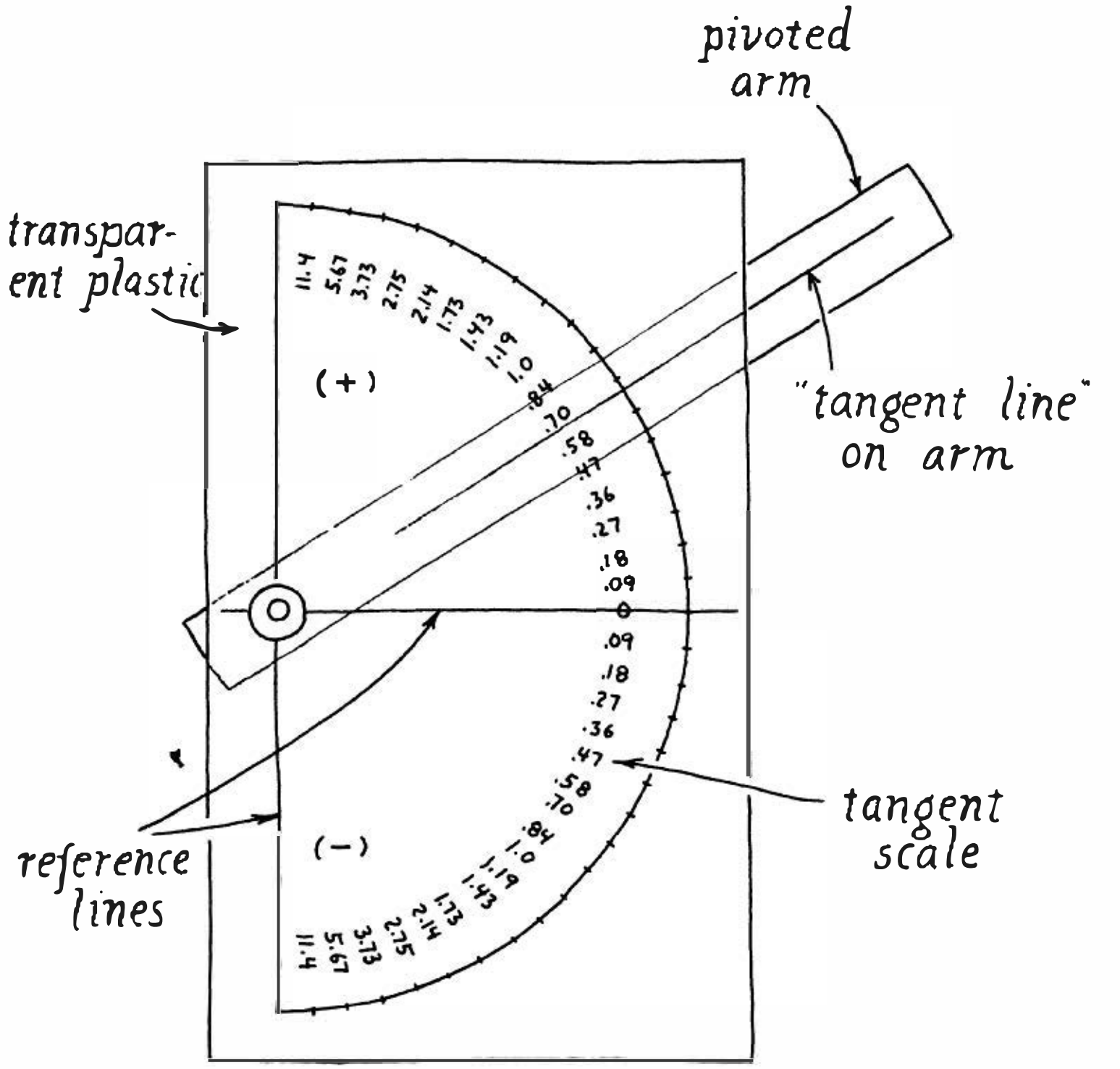


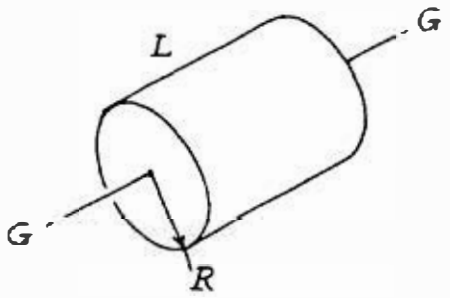
Fig. A-8. You can make a protractor of transparent plastic film (drafting film) and calibrate it in tangents rather than in angles. A line drawn on a pivoted arm is aligned with the curve, or with a previously drawn slope line. A great time saver.

tractor to measure the angle θ whose tangent is then found in a trigonometric table (slope = tangent θ). (See Fig. A–7.)

As a third method, you can make your own protractor from drafting film, calibrating it in tangents of the angles rather than in the angle itself. I have found this a very handy tool to make when there is a great deal of graphical differentiation to be done. It is even handier if the protractor is provided with a rotating arm as in Fig. A–8. A line drawn on this arm is aligned with the slope line of the curve and the tangent of the slope angle is read directly on the protractor. The reference lines on the protractor, of course, must first be aligned with the axes of the graph.

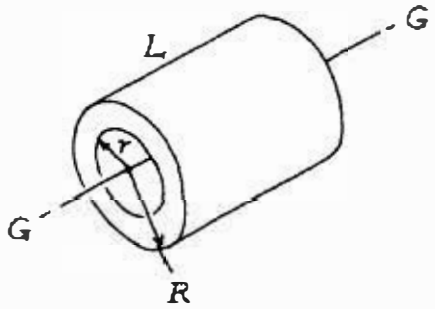
V. MOMENTS OF INERTIA

The moment of inertia of a body is, of course, a measure of its resistance to angular acceleration.



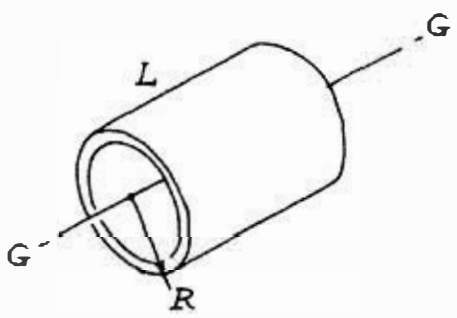
$I_G = \frac{m}{2} R^2$

$m = \rho \pi L R^2$



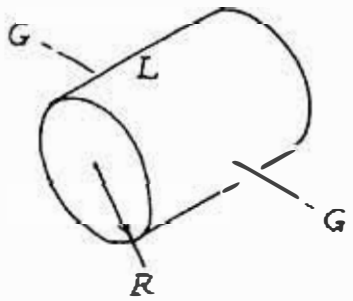
$I_G = \frac{m}{2}(R^2 + r^2)$

$m = \rho \pi (R^2 - r^2) L$



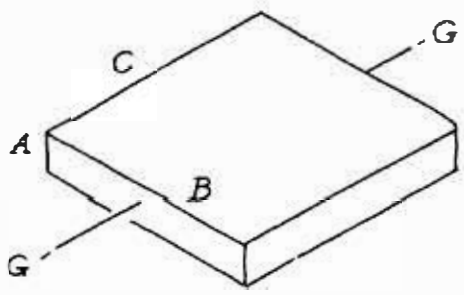
$I_G = m R^2$

$m = \rho \pi (R^2 - r^2) L$



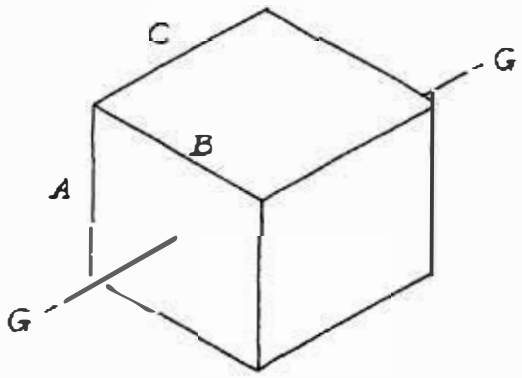
$I_G = m\left[\frac{L^2}{12} + \frac{R^2}{4}\right]$

$m = \rho \pi L R^2$



$I_G = \frac{m}{12} B^2$

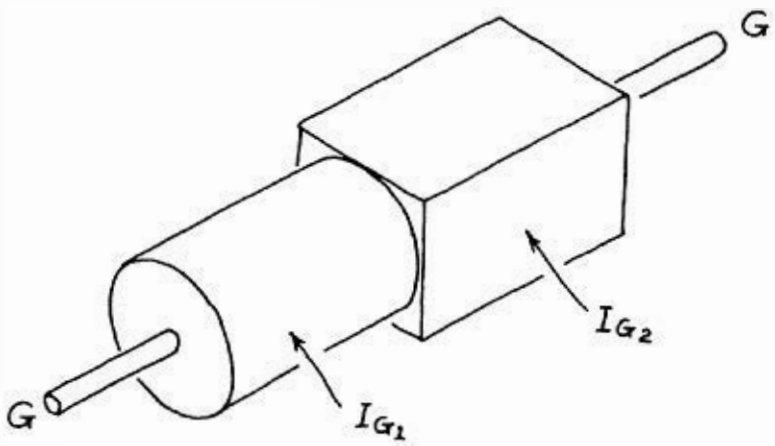
$m = \rho A B C$



$I_G = \frac{m}{12}(A^2 + B^2)$

$m = \rho A B C$

Fig. A-9. Moments of inertia for a few standard shapes. In each case the equation is for the moment about axis G-G.



$$I_{total} = I_{G_1} + I_{G_2}$$

Fig. A-10. Adding inertias of simple shapes to determine the moment of inertia of a complex body. All shapes rotating about the same axis (G-G).

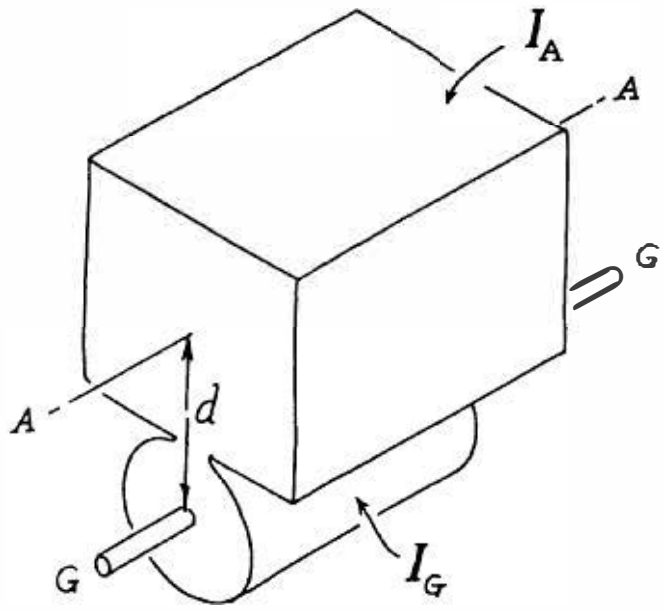
Moment of inertia is a function of the mass of a body and the distribution of this mass. In general, the greater the mass, the greater the moment of inertia; and the farther this mass is located from the center of rotation of the body, the greater the moment of inertia.

Figure A-9 shows several standard machine shapes and the equations for the inertias of these various bodies in terms of material density and body dimensions. More complete tables of this type are available in most engineering handbooks.

Figure A-9 shows the moments of inertia of elementary geometrical shapes only; however, most machine bodies are more complex. To obtain the inertia of a "compound" shape when all portions of the body are rotating about the same axis, the inertias of each portion are merely added together, as shown in Fig. A-10. If it is desired to know the

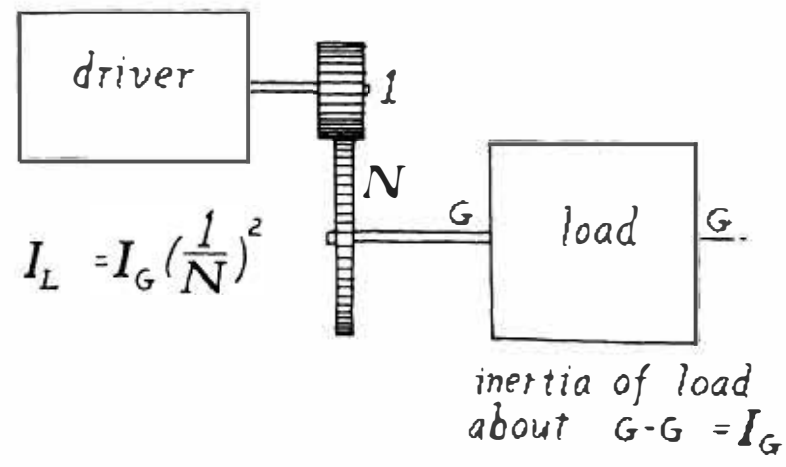
moment of inertia of a body about some parallel axis which does not pass through the body, the procedure and equation shown in Fig. A-11 is used. If a complex body consists of one shape on the axis of rotation and another one fastened to it that is off the axis of rotation, we use the off-axis formula to compute the inertia of the second body before adding it to the first.

Frequently, we need to relate the inertias of several bodies to a single point in the machine, even though these bodies are separated from the point of interest (perhaps the drive shaft of the motor) by



$$I_{total} = I_G + (I_A + m_A d^2)$$

Fig. A-11. Determining total moment of inertia of a complex body when portions of the body are centered on a parallel axis. Note that these "off axis" bodies are fastened to the cylinder and rotate with it, however, about axis (G–G). I_A equals the moment of inertia of the block about its own axis (A–A). M is the mass of the block.



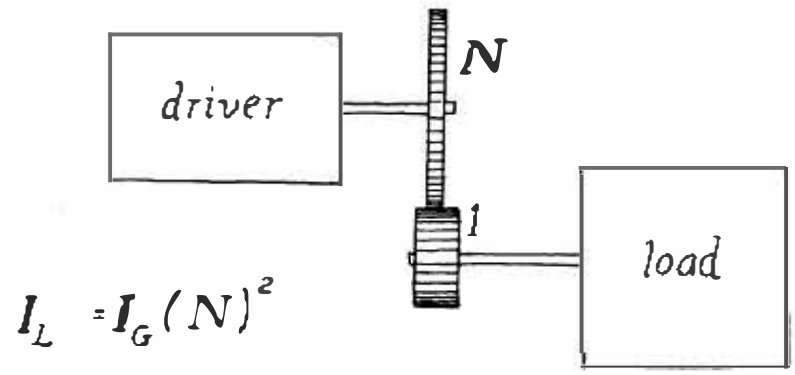


Fig. A-12. The effect of gearing on moment of inertia. In each case the calculation is for the inertia of the load as seen by the driver.

gearing, etc. In this case, the remote inertias must be multiplied by the square of the gear ratio between load and driver if the body is moving faster than the driver. If the body is moving slower than the driver, we divide by the square of the gear ratio as shown in Fig. A–12.

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